AD A278 747

NAVSHIPS 900,190

GUIDE MANUAL OF COOLING METHODS FOR ELECTRONIC EQUIPMENT

BUREAU OF SHIPS
NAVY DEPARTMENT
WASHINGTON 25, D. C.

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Washington 25, D. C. - Price $2.00~~

Navy Department
Bureau of Ships
31 March 1955

"Guide Manual of Cooling Methods for Electronic Equipment", NAVSHIPS 900,190, was originally published as Cornell Aeronautical Laboratory, Incorporated, report Number HF-710-D-16, dated 1 April 1954, under Bureau of Ships Contract NObsr-49228. This publication is the third of a series on the subject of heat transfer in electronic equipment.

The first of the series, Number HF-710-D-10, "Survey Report of the State of the Art of Heat Transfer in Miniaturized Electronic Equipment", dated 3 March 1952, Bureau of Ships Contract NObsr-49228, Cornell Aeronautical Laboratory, Incorporated, is available in the publications supply system under NAVSHIPS 900,189.

The second of the series, Number HF-845-D-2, "Manual of Standard Temperature Measuring Techniques, Units, and Terminology for Miniaturized Electronic Equipment", dated 1 June 1953, Bureau of Ships Contract NObsr-63043, Cornell Aeronautical Laboratory, Incorporated, is available in the publications supply system under NAVSHIPS 900,187.

This manual serves as a guide to the designers of electronic equipment, pointing out heat dissipation characteristics and problems, and showing how to determine the best heat-transfer methods. It is written in such a way as to be of maximum usefulness to the electronics person having a minimum of knowledge about thermodynamics.

This publication, in part or in entirety, may be used to facilitate the preparation of other government publications.

Requests for additional copies of this publication should be directed to the nearest District Publications and Printing Office. Commercial firms must submit requests to the local Navy Inspector.

B. E. Manseau

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Rear Admiral, USN
Acting Chief, Bureau of Ships

TABLE OF CONTENTS

		Page
I.	PROMULGATING LETTER	
II.	INTRODUCTION	2
III.	BASIC DESIGN CONCEPTS	
	A. FUNDAMENTAL PRINCIPLES OF HEAT REMOVAL	5
	B. APPROACHES TO THE THERMAL DESIGN OF ELECTRONIC EQUIPMENT	7
	C. METHODS OF THERMALLY RATING ELECTRONIC EQUIPMENT AND PARTS	7
IV.	INTRODUCTION TO HEAT TRANSFER	
	A. GENERAL	9
	B. CONDUCTION	9
	C. CONVECTION	10
	D. EVAPORATION AND CONDENSATION	11
	E. RADIATION	11
	F. COMBINED MODES OF HEAT TRANSFER	13
	G. RESISTANCE CONCEPT FOR HEAT FLOW THROUGH A WALL	13
	H. THE ULTIMATE SINK	15
	J. RELATIVE MAGNITUDES OF HEAT TRANSFER PROCESSES	15
V.	NATURAL METHODS OF HEAT REMOVAL	
	A. GENERAL	17
	B. THEORY	17
	1. Heat Transfer by Metallic Conduction	17
	2. Free Convection in Gases	28
	3. Radiation	34
	4. Radiation and Gaseous Conduction	39

TABLE OF CONTENTS (Cont'd.)

	Page
C. NATURAL METHODS OF COOLING ELECTRONIC PARTS	
1. General	41
2. Vacuum Tubes	41
a. Hot Spot Locations	41
b. Shields for Removing Heat from Tube Envelopes	41
(1) General	41
(2) Miniature Tube Shields	43
(3) Subminiature Tube Shields	47
c. Unshielded Tubes	61
3. Resistors	61
4. Iron Core Inductors	63
D. NATURAL METHODS OF COOLING ASSEMBLIES	63
1. Metallic Conduction Cooling	63
2. Plastic Embedment	69
E. THE PLACEMENT OF PARTS WITHIN SUBASSEMBLIES	70
F. NATURAL METHODS OF COOLING ELECTRONIC EQUIPMENT CASES	70
VI. FORCED AIR COOLING	
A. GENERAL THEORY	74
B. FORCED AIR COOLING DESIGN	81
C. PRESSURE DROP AND COOLING POWER REQUIREMENTS	84
D. FANS AND BLOWERS	86
E. COLD PLATE HEAT EXCHANGERS	89

TABLE OF CONTENTS (Cont'd.)

		Page
VII.	LIQUID COOLING	
	A. GENERAL	93
	B. THEORY	94
	C. DIRECT LIQUID IMMERSION	101
	1. General	101
	2. Design Considerations	103
	D. DIRECT LIQUID IMMERSION WITH AGITATION	108
	E. DIRECT FORCED LIQUID COOLING	108
	1. General	108
	2. Typical System	108
	3. Direct Spray Cooling	111
	4. Detailed Design Considerations	113
	F. INDIRECT FORCED LIQUID COOLING SYSTEMS	116
	1. General	116
	2. Liquid Cooled Cold Plates	117
	3. Improvement of Existing Equipment by Liquid Cooling	117
	4. New Equipment	117
	G. COMPOSITE LIQUID COOLING SYSTEMS	120
	H. CHARACTERISTICS OF COOLANTS	120
VIII.	VAPORIZATION COOLING	
	A. GENERAL	121
	B. NOTES ON VAPORIZATION COOLING AND BOILING	122
	C. DIRECT VAPORIZATION COOLING SYSTEMS	123
	1. Liquid Potting	123
	2. Expendable Direct Vaporization Cooling	125

TABLE OF CONTENTS (Cont'd.)

	Page
3. Direct Spray Systems	128
D. INDIRECT VAPORIZATION COOLING	128
1. General	128
2. Examples of Indirect Vaporization Cooling Systems	130
E. DESIGN NOTES	133
F. NOTES ON PROPERTIES OF VAPORIZATION COOLANTS	134
1. Freons	134
2. Perfluorocarbons	134
3. Fluorochemicals	136
4. Other Coolants	137
IX. THE SELECTION OF OPTIMUM COOLING METHODS	
A. GENERAL	138
B. HEAT TRANSFER WITHIN A UNIT	138
C. HEAT TRANSFER TO THE ULTIMATE SINK	142
D. DESIGN EXAMPLE OF THE SELECTION OF OPTIMUM COOLING METHODS	143
X. THE TEMPERATURE LIMITS AND RATINGS OF ELECTRONIC EQUIPMENT AND PARTS	
A. GENERAL	145
B. THE THERMAL LIMITATIONS OF VACUUM TUBES	145
1. General	145
2. Heat Transfer Within Vacuum Tubes	146
3. Limiting Factors	147
4. Vacuum Tube Ratings	149

TABLE OF CONTENTS (Contd.)

		Page
C.	THE THERMAL LIMITATIONS OF SEMICONDUCTOR DEVICES	151
D.	THE THERMAL LIMITATIONS OF MAGNETIC CORE DEVICES	156
E.	THE THERMAL LIMITATIONS OF RESISTORS	159
	1. General	159
	2. Fixed Resistors	159
	3. Variable Resistors	160
F.	THE THERMAL LIMITATIONS OF CAPACITORS	161
	1. General	161
	2. Fixed Capacitors	161
	3. Variable Capacitors	162
XI.	RECOMMENDATIONS FOR DETERMINING AND IMPROVING THE THERMAL PERFORMANCE OF ELECTRONIC EQUIPMENT	
A.	THE THERMAL ANALYSIS OF ELECTRONIC EQUIPMENT	163
B.	IMPROVING THE THERMAL PERFORMANCE OF EXISTING ELECTRONIC EQUIPMENT	163
C.	THE DESIGN OF EFFICIENT ELECTRONIC CIRCUITS	165
XII.	APPENDIX	
A.	SYMBOLS AND NOMENCLATURE	169
B.	LIST OF ASSOCIATED REPORTS	172
C.	TABLES AND CHARTS	173
D.	BIBLIOGRAPHY	195

LIST OF ILLUSTRATIONS

Figure No.		Page
1.	Temperature Gradient through a Wall	13
2.	Electrical Circuit Analogy for Heat Flow through a Wall	14
3.	Conduction through a Single Plane Wall or Bar Insulated on the Sides	18
4.	Conduction through a Bar	19
5.	Conduction through a Cylinder	19
6.	Conduction through Composite Walls	20
7.	Contact Resistance as a Function of Contact Pressure	23
8.& 9.	Thermal Conductivity as a Function of Contact Pressure	24
10.	Composite Bar with Contact Resistance	26
11.	Chart for Calculation of Free Convective Heat Dissipation in Air	Envelope
12.	Chart for Calculation of Radiant Heat Dissipation	"
13.	JAN Tube Shield	44
14.	Conduction Tube Shield	46
15.	Radiation - Conduction - Convection Shield	48
16.	Ives' Shield	49
17.	Wrap-around Subminiature Tube Shield	51
18.	Fuse Clip Type Subminiature Tube Shield	52
19.	Several Types of Subminiature Tube Shields	53
20.	Drilled Metal Block for Mounting Subminiature Tubes	54
21.	Typical Gradients on a Wrap-around Tube Shield	56
22.	Summary of Subminiature Tube Shield Evaluation	57
23.	Shield Configurations	60

LIST OF ILLUSTRATIONS (Contd.)

Figure No.		Page
24.	Subchassis Mount using a "Sink Strainer"	62
25.	Conventional and Conduction Cooled Transformers	64
26.	Video Amplifier Subassembly	65
27.	Drilled Metal Block Construction	67
28.	Construction Methods	68
29.	Air Space Conductance Curves for Group Mounting of Resistors	71
30.	Recommended Construction Practices in Conventional Equipment	72
31.	Temperature Rise and Power Dissipation of Typical Etched Aluminum Cases	73
32.	Tube Bundles with Tubes in Line and Staggered	77
33.	Typical Centrifugal Fan	87
34.	Propeller-type Fan	87
35.	Axial Flow Fan	87
36a.	Characteristic Curve of a Centrifugal Fan at Constant Speed	88
36b.	Fan and Duct System Performance	88
37.	Cross Section through a Typical Forced Air Cooled Cold Plate	91
38.	Equivalent Electrical Circuit	95
39.	Heat Transfer Diagram of a Simple Direct Liquid Cooled Subassembly	102
40.	Observation of the Heat Distribution of a Subminiature Tube	107
41.	Direct Liquid Cooling with Agitation	109
42.	Direct Forced Liquid Cooling System	110
43.	Direct Spray Cooling System	112

LIST OF ILLUSTRATIONS (Contd.)

Figure No.		Page
44.	Liquid Cooled Cold Plate	118
45.	Vaporization Cooled Liquid Potted Subassembly	124
46.	Typical Direct Expendable Vaporization Cooled Electronic Device	126
47.	Direct Evaporative Spray Cooling System	129
48.	Liquid to Liquid Vaporization Cooling System	131
49.	Liquid to Liquid Indirect Vaporization Cooling System	132
50.	Comparison of Methods of Cooling	140
51.	Miniature Tubes Painted with Thermocolor and Tempilac	148
52.	Relative Temperatures of Typical Tubes	152
53.	Bulb Temperature vs. Ambient Temperature, 6AK5 vs. 5702	153
54.	Selenium Rectifier Ratings	155
55.	The Thermal Power of Subminiature Tube Circuits	167
56.	Pressure Temperature Relationshio of "Freon" compounds	190
57.	Fluorochemical N-43 Vapor Pressure - Temperature	191
58.	Fluorochemical N-43 Temperature vs. Density - Viscosity	192
59.	Fluorochemical 0-75 Vapor Pressure - Temperature	193
60.	Fluorochemical 0-75 Temperature vs. Density - Viscosity	194

LIST OF TABLES

Table No.		Page
1.	Representative Magnitude of Heat Transfer Processes	16
2.	Comparison of Thermal Conductance Measurements at 10 psi.	25
3.	Engineering System of Units for Free and Forced Convection	29
4.	Significant Dimension "L"	30
5.	Convective Cooling Constants	31
6.	Constants for Use in Free Convection Equations for Round Cylinders	76
7.	Reynolds' Number Functions for Cylinders of Square Cross Section	76
8.	Reynolds' Number Functions for Staggered Tubes	78
9.	Ratio of Mean Film Coefficient for N Rows Deep to that for 10 Rows Deep	79
10.	Comparison of Heat Transfer Data for Air, Silicone Fluid, Transformer Oil and Freon in Multitube Apparatus.	101
11.	Resistivity and Dielectric Strength of Fluorochemicals	137
12.	Unit Heat Dissipations of Typical Tube Types	149
13.	Typical Bulb Temperatures at 23°C Ambient Temperature	150
14.	Bulb Temperatures at Various Ambient Temperatures	150
15.	Heat Conduction Data for Various Materials at approximately 65°C	173
16.	Physical Properties of the Useful Metals	174
17.	Temperature Conversions	175
18.	Properties of Air	176
19.	Properties of Air, Helium, Hydrogen and Nitrogen at 100°C.	177
20.	Emissivity Values of Common Materials	178

LIST OF TABLES (Contd.)

Table No.		Page
21.	Total Emissivity Values for Various Metals and Glasses	179
22.	Properties of Water	180
23.	Properties of DC-550 Silicone Fluid	181
24.	Properties of DC-701 Silicone Fluid	182
25.	Properties of DC-200 Silicone Fluid	183
26.	Properties of Mineral Oil	184
27.	Typical Properties of Perfluorocarbon Liquids	185
28.	Typical Solubility Properties of Perfluorocarbon Liquids	186
29.	Properties of Fluorochemical N-43	187
30.	Properties of Fluorochemical 0-75	188
31.	Coolant Properties	189

LIST OF EXAMPLE PROBLEMS

Example Number	Description	Page No.
1	Calculation of conduction through a bar	19
2	Calculation of conduction with and without contact resistance	26
3	Calculation of convective cooling	32
4	Calculation of cooling by radiation	36
5	Calculation of forced convection cooling	81
6	Calculation of direct liquid cooling	96
7	Computation for an expendable vaporization cooling system	125
8	Calculation of the selection of optimum cooling methods	143

II. INTRODUCTION

The military requirements for electronic equipment have been steadily increasing. This trend in demand for equipment having improved performance, decreased size, greater reliability and complexity of function is continuing. For example, a typical destroyer in 1937 incorporated a total of approximately 60 active vacuum tubes maintained by a single technician. By 1944 a destroyer utilized 850 tubes. In 1952 this total had increased to 3200 tubes, serviced by 14 technicians. The trend to increased complexity will not change in the foreseeable future. As a result, electronic design problems are multiplying in severity. Production mechanization and electronic cooling programs are among the current endeavors to obtain satisfactory economy and reliability. It is necessary that these and associated problems be resolved in order to achieve acceptable electronic devices.

Reliability is of paramount importance. Reliability has been defined as the ratio of the time the equipment is in usable operating condition to the total time the equipment is required for use. In general, the reliability of current electronic equipment is poor. In fact, it has been concluded that it would be economical to pay more than twice the present cost for military electronic equipment if reliability could be improved by 50 per cent.

Miniaturization has led to ever increasing heat concentrations with the necessity for adequate heat transfer within equipment. Effective heat removal is of prime importance in obtaining satisfactory life, reliability and electronic performance. If the temperatures of electronic parts exceed certain values, malfunctioning and failures follow. Thus, the science of heat transfer must be employed in electronic design.

In order to establish a firm foundation of knowledge, the initial phase of this program was a national survey of the state of the art of heat transfer in electronic equipment. The findings of this effort are presented in Cornell Aeronautical Laboratory Report No. HF-710-D-10. It was found that the electronic circuit design of most military electronic equipment has been excellent. Unfortunately, the mechanical and thermal designs have not been as satisfactory. Acceptable reliability can only be achieved if the electronic, thermal and mechanical designs are all well executed. Thus, the thermal design is equally as important as the circuit design. An investment of effort on the thermal design will return good interest in terms of improved reliability.

This Manual has been prepared to assist the electronic engineer in the thermal design of miniaturized equipment. In effect, this Manual sorts and places the known electronic cooling design information together between two covers in a brief, predigested form for use by electronic engineers. It has been deliberately written at an appropriate technical level so that engineers without heat transfer backgrounds can design acceptable equipment. Therefore, this Manual outlines thermal parameters which, it is believed, will temper the designer's judgment and lead to the best practical solution. Design formulae contained herein are usually simplified approximations. Frequently, qualitative, rather than quantitative, information is presented. When specific recommendations are given, care must be taken in adopting such recommendations so that they do not conflict with the requirements of the contracting activity.

Two groups of recommendations are incorporated herein. The first series are based upon the concept that improved reliability can be achieved with existing equipment provided it can be modified for better cooling. For this reason, several sections of the discussion are devoted to improvement of present equipments. The second group of recommendations are for the new equipment designers. These data are not only for those designing equipment of the future, but for those now designing equipment compromised by the temperature limited present day parts.

This Manual emphasizes ground-based and shipboard electronic equipment. No consideration has been given to non-steady state heat transfer or operation at environmental air pressures significantly less than one atmosphere. The Ohio State University Research Foundation has prepared a series of reports related to transients and more detailed analysis of these phases of the problem. Further, for the purposes of the Manual, it has been assumed that unitized construction will be generally used, the equipment being composed of subassemblies and assemblies.

The thermal design of electronic gear is a relatively new science. In general, due to the current limited knowledge, only reasonable design approximations can be made. Much remains to be accomplished in this relatively new field. There is a need for the standardization of cooling means and systems. Also, widely accepted standard heat transfer liquids should be developed. If the operating temperatures of electronic parts can be increased, some of the current thermal problems will be alleviated. Further, present methods of temperature rating should be improved. The use of thermal environment or surface temperature rating, in lieu of ambient temperature rating, will aid in this matter. In addition, techniques should be developed to obtain an analytical method of determining the rating and limits of each equipment type under given environmental conditions.

We wish to acknowledge the excellent cooperation we have received during this program from government agencies, universities and industries. References of souroe material are listed in the bibliography, together with reference numbers at pertinent locations in the discussion. The terminology used herein is presented in Cornell Aeronautical Laboratory Report No. HF-845-D-2. Symbols are incorporated in Appendix A.

Work on heat transfer in electronic equipment is continuing at this Laboratory. Future publications will supplement this Manual. Appendix B consists of a summary of reports already written and also planned under this program. Detailed information in these matters can be obtained from Mr. James W. Brush, Code 818C, Bureau of Ships, Washington 25, D.C. Comments are solicited.

This Manual is not to be construed as an endorsement of any commercial products mentioned.

III. BASIC DESIGN CONCEPTS

A. FUNDAMENTAL PRINCIPLES AND THEIR APPLICATION TO ELECTRONIC HEAT REMOVAL

Electronic equipment must incorporate means for adequate heat rejection in order to provide reliable performance at thermal equilibrium. This will be necessary until the efficiency of electronic equipment becomes of high order. As long as power is dissipated it will be rejected in the form of heat. The purpose of any electronic cooling system is to provide a low resistance thermal path to a heat sink to absorb this waste heat. When used in conjunction with a heat sink at a reasonably low temperature, such a system will reduce the temperature rise of electronic parts and equipment.

There are several important factors which must be thoroughly understood prior to the initiation of the design of any electronic equipment.

1. Temperature difference controls the rate of heat transfer in any given configuration.

2. At thermal equilibrium a heat balance is always maintained. The natural law of the conservation of energy applies and all of the heat generated will be rejected, if necessary, by means of high temperature gradients.

3. In order to define the thermal parameters, the electronic engineer must first determine temperature and rate of heat production. The dissipated power can usually be measured. Even though temperature measurement is not as easy, the determination of temperature is necessary, since it is the only other measurable quantity in the thermal circuit. Further, temperature is a measure of the quality of the thermal design.

4. The electronic designer, in determining his heat transfer system, must first select the most simple and economical cooling system applicable to the proposed design, environment and specifications. Many factors must be considered: space, economy, power to operate the cooling system, the temperature limitations of the electronic parts, the circuit configuration, the thermal environment, the heat concentration and the ultimate sink. A satisfactory thermal design should start on the drafting board simultaneously with the electrical and mechanical design activities.

5. The thermal design must be such that the electronic performance is not significantly affected. In certain instances the optimum cooling technique may not be consistent with electronic performance. When this situation arises, it is recommended that alternate cooling means or circuit designs be utilized. Compromise designs are more often the rule than the exception.

6. Heat transfer design inherently includes reasonably wide tolerances. Due to the nature of heat transfer, a high degree of design accuracy can seldom be achieved. However, this does not impede the design of a practical cooling system. Once the type of cooling system has been tentatively selected, the thermal analysis must be made. The thermal circuit should be approximated as closely as possible to permit mathematical analysis.

B. APPROACHES TO THE THERMAL DESIGN OF ELECTRONIC EQUIPMENT

There are two basic approaches to the design of electronic equipment with satisfactory thermal performance:

1. The "brute force" method, wherein high temperature electronic parts are used to permit operation without special cooling means, can be used to provide heat rejection through the operation of parts at high temperatures. When used at low ambient temperatures to alleviate the effects of excessive hot spots, this approach is inherently inefficient and expensive. High temperature electronic parts should only be used for operation in environments with high ambient temperatures and in conjunction with an adequate heat removal means. The utilization of high temperature parts is not recommended as a remedy for the deficiencies of an inferior heat removal system.

2. The controlled heat removal method should be used in all Military electronic equipment. This approach embodies the most effective methods of heat transfer and includes special techniques. It requires the application of careful design of the entire thermal system and the establishment of low temperature gradients to protect temperature sensitive parts and circuits. Of prime importance is the necessity of directing the heat from the sources along specified paths to a low temperature sink so that the heat is not indiscriminately scattered and transferred into adjacent electronic parts. The magnitude and location of heat flow must be controlled. For these reasons, certain heat removal methods which will cool heat sources and transfer their heat into other parts are considered undesirable.

C. METHODS OF THERMALLY RATING ELECTRONIC EQUIPMENT AND PARTS

In general, electronic parts are individually rated for certain performance at specified ambient temperatures. Ambient temperature rating is rapidly becoming obsolete because it is indeterminate. Almost every organization has a slightly different definition and interpretation of ambient rating. Such rating is, in some respects, almost worthless, and thermal performance "loopholes" are provided through its use. Not only should individual part rating be considered, but also the group characteristics must be assayed in order to avoid shifts in values or outright failure. The thermal interaction due to mutual heating and cooling can cause greatly increased temperatures over those obtained with solitary parts.

Ambient temperature is only the temperature of the medium surrounding an object. This does not define the true thermal situation as it may exist around an electronic part. With densely packaged equipment, the local air temperature is not directly related to the heat radiation or conduction effects from nearby heat sources. These effects are frequently significant and can lead to the overheating of parts even though the ambient rating is not exceeded. Ambient temperature rating sufficed for the conventional World War II type of equipment, since widely separated heat sources were operated at relatively low temperatures.

The primary function of any method of part rating is to determine the thermal state of the internal functioning elements. The limiting temperatures are those which the constituents can withstand before they oxidize, melt, decompose or change value. Unfortunately, internal temperatures are usually difficult to measure. Therefore, the best practical indication of the thermal index of the inside of an electronic part is its surface temperature or the change in value of a readily measurable electrical parameter. Neither of these characteristics are necessarily related to the temperature of the surrounding air.

Like electronic parts, most electronic equipment is usually rated in terms of ambient temperature. Military specifications also incorporate ambient ratings. It is believed that military specifications and equipment ratings should be modified to incorporate thermal environment ratings which can provide the equipment designer and user with definite thermal parameters. The thermal environment can be defined as the condition of (1) fluid type, temperature, pressure and velocity; (2) surface temperatures, configurations and emissivities and (3) all conductive thermal paths surrounding an electronic device. The ambient temperature is only the fluid temperature surrounding an electronic device. It is one of the factors contributing to the thermal environment.

Military specifications for electronic equipment should provide and define facilities for the removal of the dissipated heat at the installation location. The thermal situation should be described both before and after the installation of the equipment.

The electronic equipment manufacturers should furnish the user with the heat dissipation rate of each unit of equipment, the cooling requirements, the maximum temperature of the equipment at specified locations which are indicative of the thermal situation and, if a coolant is used, other pertinent information such as pressure drop and temperature gradients.

IV. INTRODUCTION TO HEAT TRANSFER

A. GENERAL

This section is a brief review of the modes and basic laws of heat transfer. Its purpose is to serve only as an introduction to re-acquaint the electronic engineer with the basic principles involved.

Heat or thermal energy is transferred from one region to another by virtue of temperature difference. The two fundamental axioms are that heat flows only from a high temperature region to one of lower temperature, and that the heat emitted by the high temperature region must be exactly equal to that absorbed by the low temperature region.

When heat is transferred at a steady rate and the temperature at any given point is constant, the steady state is said to exist. On the other hand, if the heat flow is a function of time, the flow is said to be in the unsteady state. An example of the latter is the warm-up period of an electronic assembly. This Manual does not consider such transient conditions but only those occurring after thermal equilibrium has been reached.

In general, there are three modes or methods of heat transfer: conduction, convection and radiation. They may occur singly or simultaneously. While evaporation and condensation may be classified under convection, they are usually considered separately since mass transfer, as well as heat transfer, occurs.

B. CONDUCTION

Heat conduction is considered to be caused through molecular oscillations in solids and elastic impact in liquids and gases. The basic law of heat conduction in the steady state and in its most simple form (heat transfer through a wall) is:

$$q = k \frac{A}{L} \Delta t \qquad (1)$$

where:

- q is the rate of heat transfer
- k is the thermal conductivity of the material
- A is the cross-sectional area perpendicular to the direction of heat flow
- L is the length of heat flow path
- Δt is the temperature difference causing the heat flow

Heat flow is analogous to Ohm's Law. Rewriting equation (1) as:

$$q = \frac{\Delta t}{\frac{L}{kA}} \qquad (2)$$

it can be seen that q is analogous to I, Δt to E, and L/kA to R.

The thermal conductivity k is the quantity of heat which will flow across unit area in unit time when the length of heat path is unity and the temperature gradient across this path is unity. Its numerical value depends on the material, being high for metals and low for insulators. For example, the thermal conductivity of copper is over 300 times that of glass.

C. CONVECTION

The process of heat transfer from the surface of a solid to moving masses of fluids, either gaseous or liquid, is known as convection. This mode of heat transfer is brought about mainly through circulation of the fluid. For example, the surface of a warm object situated in still air at a lower temperature heats the air adjacent to the surface. The heated air becomes less dense as its temperature increases and induces convection currents. When the circulation is caused only by differences in density, the process is called natural or free convection. Circulation may be forced mechanically by blowers, pumps, etc., in which case the heat transfer is called forced convection.

The mechanism of convection may be explained by considering a cool stream of air flowing past a heated surface. Immediately adjacent to the surface there exists a film of air varying in velocity from zero at the surface to the velocity of the main stream at its outer side. This film offers a resistance to heat flow and is influenced by the nature of the flow. In free convection the film is usually in laminar or streamline motion and relatively thick, causing high resistance to heat flow. However, in forced convection, the higher velocity tends to decrease the thickness of this film improving the heat transfer. In laminar flow the film moves in streamline motion, while in turbulent flow, two not sharply defined layers are believed to exist, the inner in streamline and the outer in turbulent motion. Heat is believed to pass through the streamline layer mainly by conduction and through the turbulent layer by mixing and diffusion with the fluid in the main stream.

The basic equation for convection is :

$$q = h_c A \Delta t \qquad (3)$$

where:

q is the heat transfer rate,

h_c is a convection coefficient of heat transfer

A is the surface area

Δt is the temperature difference between the surface and the main fluid stream

The value of h_c is influenced by many factors including not only the properties of the fluid such as viscosity, density, etc., but the flow conditions and surface characteristics as well. The resistance concept may also be applied to convection in which case the term $1/Ah_c$ is the thermal resistance.

D. EVAPORATION AND CONDENSATION

Evaporation and condensation are characterized by a change of state involving a liquid evaporating to the vapor state and a vapor condensing to the liquid state, respectively. The basic equation for these processes is the same as that for convection.

Evaporation is a general term used whenever molecules leave the surface of a liquid changing to the vapor state. The amount of heat necessary to evaporate a unit mass of a liquid to a vapor is called the heat of vaporization. The process may occur with or without boiling. For example, water will evaporate into air by diffusion if the vapor pressure of the water is greater than the partial pressure of the water vapor in the air. Evaporation can be speeded up by heating the water. If the temperature is increased until the vapor pressure of the water equals the ambient atmospheric pressure, boiling will take place. Evaporation usually involves high rates of heat transfer and the coefficients may be as much as 200 times that for air in forced convection.

Condensation occurs when a vapor condenses to a liquid on a cooler surface. Here the heat of vaporization is now released. Like vaporization, high rates of heat transfer are usually associated with condensation.

E. RADIATION

Bodies under thermal agitation induced by temperature emit thermal radiation in the form of electromagnetic waves ranging in wave length from the long infrared to the short ultraviolet. Radiation emitted from a body can travel undiminished through a vacuum or through gases with relatively little absorption. When radiation is intercepted by a second body, part may be absorbed as thermal energy, part may be reflected from the surface, and part may be transmitted still in electromagnetic wave form through the body as in the case of glass.

Consider a body in space receiving radiant energy from some source. If all the incident radiation is absorbed with zero energy being reflected or transmitted, it is a perfect absorber and called a "blackbody". There are no perfect absorbers in nature although some bodies come very close to exhibiting blackbody characteristics. The ratio of the amount of energy absorbed by an actual body to that by a thermal "blackbody" is called the "absorptivity". In the absence of conduction and convection, a body at thermal equilibrium which receives radiation must necessarily emit radiant energy equal to that absorbed. Hence a body which is a good receiver or absorber is a good radiator or emitter. The ratio of the amount of radiant energy emitted by an actual body to that emitted by the ideal blackbody is called the "emissivity" and is numerically equal to the absorptivity. Its numerical value is always less than unity. The emissivity of polished copper, for example, is 0.023, whereas that of oxidized cast iron may be as high as 0.95.

The basic equation for the radiation from a blackbody is:

$$q_b = \sigma A T^4 \tag{4}$$

where:

q is the rate of energy emitted by a blackbody

σ is the Stephan-Boltzmann constant

A is the surface area

T is the absolute Temperature

For actual bodies, equation (4) must be modified for departure from ideal blackness and, since the net exchange of radiant energy between two bodies is usually required, it must be modified depending on the geometry of the system. The general equation for the <u>net</u> rate of exchange of radiant heat between two non-black bodies is:

$$q_r = F_e F_a A\sigma(T_1^4 - T_2^4) \tag{5}$$

where:

F_e is an emissivity factor to allow for departure from black body conditions

F_a is a configuration factor based on the geometry of the system (not all of the radiation emitted by a body may be intercepted by the second body)

T_1 and T_2 are the temperatures of the hot and cold bodies respectively.

The net radiation between two bodies is thus proportional to the difference in the fourth powers of the absolute temperatures, whereas conduction and convection in general are proportional to the difference in the first powers of the temperatures.

F. COMBINED MODES OF HEAT TRANSFER

Convection and radiation may occur simultaneously to provide parallel heat flow paths. The resistance concept may be applied to radiation by considering a fictitious radiation coefficient, h_r, such that the convection and radiation coefficients are additive when a surface loses heat by both modes.

The radiation coefficient is defined as:

$$h_r = \frac{q_r}{A\Delta t} \tag{6}$$

where Δt is the difference in surface and air temperatures.

Thus, the total heat transferred from a surface is:

$$q_{\text{total}} = q_{\text{convection}} + q_{\text{radiation}} = (h_c + h_r)\, A\Delta t \tag{7}$$

An electronic box situated in a lower temperature environment might lose heat by both convection and radiation from its outer surfaces.

G RESISTANCE CONCEPT FOR HEAT FLOW THROUGH A WALL

Consider a wall as in Fig. 1 on one side of which is air at a higher temperature than the air on the opposite side.

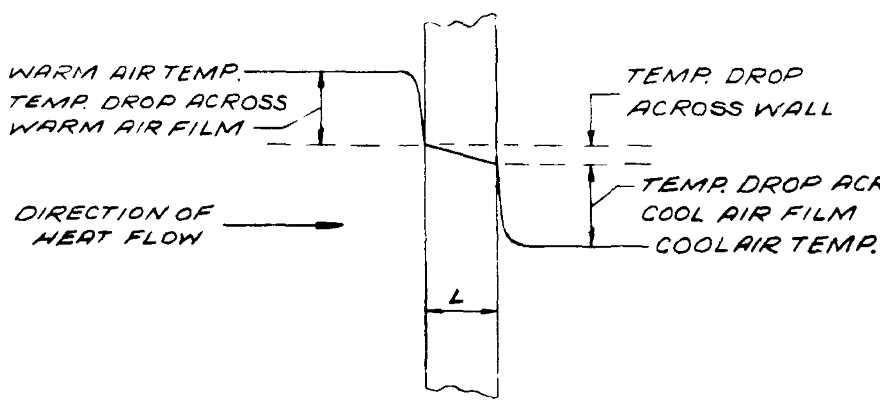


Fig. 1. Temperature Gradient Through A Wall

Neglecting radiation in this simple example, Fig. 1 shows the temperature gradient from the warm air to the cool air. Heat is transferred by convection across the air film, then by conduction across the wall, and finally by convection to the cooler air. Since this is a thermal series circuit involving three resistances, the equation for the heat flow would be:

$$q = \frac{\Delta t_{total}}{\frac{1}{Ah_c} + \frac{L}{Ak} + \frac{1}{Ah_c}} = \frac{\Delta t_{total}}{\Sigma R} \qquad (8)$$

where:

Δt_{total} is the difference between the warm and cool air temperatures, and

h_c and $h_c{}'$ are the convection coefficients of the warm and cool air films respectively.

Each term in the denominator of equation (8) represents a thermal resistance analogous to the resistances in a simple series electrical circuit. This analogy is shown in Fig. 2.

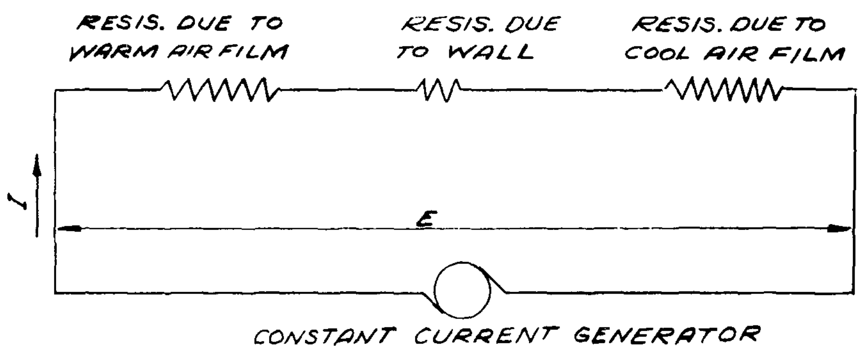


Fig. 2. Electrical Circuit Analogy for Heat Flow Through a Wall.

The temperature drop or gradient across each resistance is proportional to the resistance. Thus, the temperature drop across the wall, for example, is:

$$\Delta t_{wall} = \frac{\frac{L}{Ak}}{\Sigma R} (\Delta t_{total}) \qquad (9)$$

H. THE ULTIMATE SINK

Unlike the closed electrical circuit, the thermal circuit is an open one. The heat generated in a vacuum tube, for example, is transferred by various modes through various channels and ultimately reaches a heat sink. The cooling process is simply one of transporting thermal energy from a heat source to a heat sink at a lower temperature. The design problem is to provide a thermal path of low resistance and a low temperature sink so that the temperature of the heat sources (vacuum tubes, etc.) will not be excessive. Further, non-heat producing sources critical to temperature must be protected from overheating.

In the final analysis the ultimate sink is the earth's atmosphere, large bodies of water, or the earth itself. However, from the practical view, the electronic designer may have available intermediate sinks such as cooled air which must transport the heat from the electronic assembly to the ultimate sink. In steady-state heat transfer, it is erroneous to consider the chassis of a conventional electronic assembly as a heat sink because the chassis has finite heat capacity and heat must be removed from it at the same rate as that entering.

J. RELATIVE MAGNITUDES OF HEAT TRANSFER PROCESSES

In order to develop some conc of the relative magnitude of various heat transfer processes, Table I is presented. The values listed are representative only and may vary with conditions. Comparison is made on the basis of conductance which is the heat transfer rate per unit area per degree temperature difference. In free convection from the vertical plate, for example, the conductance is reported per degree difference in plate and air temperatures.

TABLE I

Representative Magnitude of Heat Transfer Processes

	$\frac{\text{Btu}}{(\text{hr.})(\text{sq.ft.})(^\circ\text{F})}$	$\frac{\text{Watts}}{(\text{sq.in.})\ (^\circ\text{C})}$
Conduction through copper 0.1 in. thick.	26160.	95.20
Conduction through pyrex glass 0.1 in. thick	87.36	0.322
Conduction through cork board 0.1 in. thick	3.0	0.011
Free convection from 6 in. high vertical plate at 120°C, air at 80°C.	0.96	0.00348
Forced convection, air over 6 in. plate at 8 ft./sec., mean temp. air and plate of 100°C.	2.84	0.0104
Forced convection 40°C water flowing at 5 ft./sec. in a 2 in. dia. pipe	1420.	5.19
Water boiling on a flat plate at atmospheric pressure	2000.	7.30
Steam condensing on a flat plate at atmospheric pressure	1000.	3.65
Radiation between two black bodies at 100°C and 50°C	1.72	0.0063
Radiation between two black bodies at 500°C and 50°C	7.81	0.0287

V. NATURAL METHODS OF HEAT REMOVAL

A. GENERAL

Free convection, conduction and radiation are the most common means of heat rejection within and from electronic equipment. Natural methods can be defined as those wherein heat transfer occurs without additional energy being supplied to accelerate the process. The majority of electronic equipment and parts have been designed for natural cooling in a free air environment at atmospheric pressure.

Natural methods are frequently the only practical means of removing heat from within miniaturized subassemblies. Hermetic sealing and the dense packaging of parts may prevent the utilization of, for example, forced air for internal cooling. Liquid cooling by natural means is not included in the discussion in this section. Although conduction and free convection are involved in liquid-potted equipments, it is deemed advisable to treat liquid cooling separately. Considerable emphasis is placed in this section on the cooling of vacuum tubes, because tubes produce from 70 to 85 per cent of the total waste heat of a typical equipment. Some space is devoted to other heat sources, such as resistors and reactors.

B. THEORY

1. Heat Transfer by Metallic Conduction

a. General

Since metals possess a low resistance to the passage of thermal energy, heat conduction in metals is one of the most effective modes of heat transfer. In general, it may be stated that those metals which possess low electrical resistance also possess low thermal resistance.

The basic law of heat conduction in the steady state, known as Fourier's law is

$$q = -kA \frac{dt}{dx} \tag{10}$$

where:

- k is the thermal conductivity
- A is the cross-sectional area perpendicular to the direction of heat flow
- $\frac{dt}{dx}$ is the rate of change of temperature with respect to the distance in the direction of heat flow

The minus sign is present since $\frac{dt}{dx}$ is negative

If q is measured in watts, area in square inches, length in inches, and temperature in degrees C, k will have the units of watts/(sq.in.)(^{o}C)/in. In the British system of units, using the Btu, foot, pound, ^{o}F and hour, k has the units of Btu/(hr.)(sq.ft.)(^{o}F)/ft. While k varies with temperature, the variation for metals over the range of temperature of interest in electronic cooling problems is not great. Table 15 in Appendix C presents the thermal conductivity values for various materials and also gives the comparison with yellow brass on a weight basis.

b. Conduction through a Single Plane Wall or Bar Insulated on the Sides

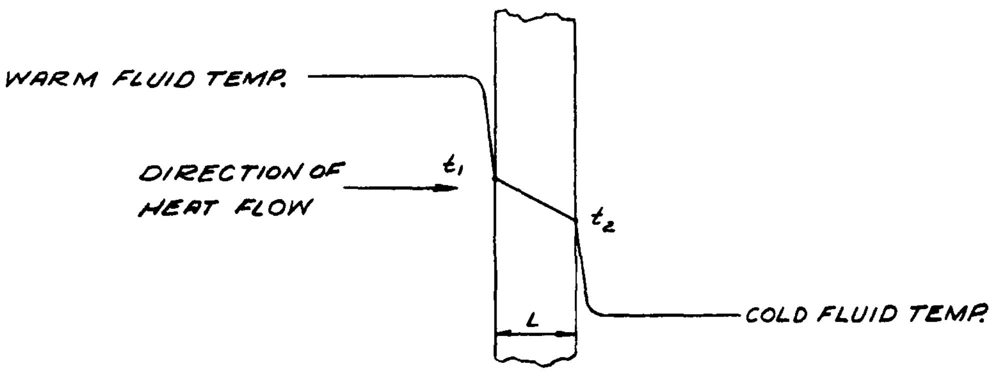


Fig. 3 Conduction through a Single Plane Wall

Figure 3 shows a plane wall, one side of which is at temperature t_1 which is higher than that of the other side, t_2. The wall is of a homogenous material with a constant thermal conductivity, k. The wall is considered to be very large so that there are no end effects, i.e., the heat flows only perpendicular to the face of the wall (unidirectional heat flow). The equation for the heat flow in this case is:

$$q = \frac{Ak\,(t_1 - t_2)}{L} \tag{11}$$

It is usual to consider k as independent of temperature. However, if k varies, a mean value of k between the temperatures should be used. It is important to note that t_1 and t_2 are the temperatures of the wall surfaces and not the temperatures of the liquids or gases which may be on either side of the wall.

Example 1. Conduction through a Bar Insulated on the Sides:

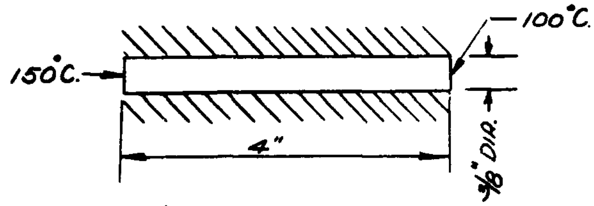


Fig. 4 Conduction through a Bar

Figure 4 shows a soft steel bar 3/8" diam., 4" long, one end of which is maintained at 150°C and the other at 100°C. The sides of the bar are perfectly insulated so that heat flows only in the direction parallel to its axis. Problem: Determine the rate of heat flow through the bar.

Solution:

$$k = 1.18 \text{ watts/(sq.in.)(}^{\circ}\text{C)/in.}$$

$$A = 1/4\,\pi\,(3/8)^2 = 0.110 \text{ sq. in.}$$

Substituting in equation (11):

$$q = \frac{0.110 \times 1.18\ (150 - 100)}{4} = 1.62 \text{ watts}$$

Note that the foregoing example is ideal in that the sides are assumed to be perfectly insulated. This situation is approximated in the case of a wall which is relatively large compared with its thickness. The more general case is that of a bar whose sides are not insulated. In this instance, both radiation and convection may cause heat to flow from the sides of the bar so that the solution becomes more complex.

c. Conduction through Cylinders and Spheres

The only other shapes which lend themselves to simple conduction calculation are the cylinder and sphere. Almost all other shapes require complicated mathematical solutions or must be solved by graphical or numerical approximations.

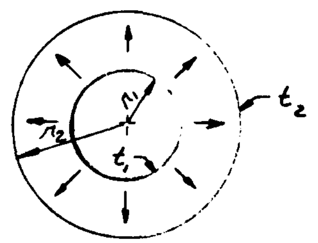


Fig. 5 Conduction through a Cylinder

Figure 5 shows a cross-section of a hollow cylinder or pipe of inner radius r_1 and outer radius of r_2. The inner and outer surface temperatures are maintained at t_1 and t_2 respectively. If t_1 is greater than t_2, heat will flow in the outward direction. It is assumed that the ratio of length to diameter is large so that end effects are negligible. The equation for the heat flow rate from the inner to outer surface <u>per inch length</u> of cylinder is:

$$q = \frac{2\pi k\,(t_1 - t_2)}{\log_e\left(\frac{r_2}{r_1}\right)} = \frac{2.72k\,(t_1 - t_2)}{\log_{10}\left(\frac{r_2}{r_1}\right)} \quad \text{watts/in.length of cylinder} \qquad (12)$$

where q is in watts, t in $^{\circ}C$, dimensions in inches, and k in watts/(sq.in.)($^{\circ}C$)/in.

If the Btu-foot-pound-hour-$^{\circ}F$ system of units is used, then q is the heat transfer rate in Btu/hr. <u>per foot length</u> of cylinder.

In the case of a hollow sphere of inner and outer radii of r_1 and r_2 respectively with corresponding surface temperatures of t_1 and t_2, the total heat conducted to the outer surface is given by

$$q = \frac{4\pi k\, r_1\, r_2\,(t_1 - t_2)}{r_2 - r_1} \qquad (13)$$

d. <u>Conduction through Composite Walls</u>

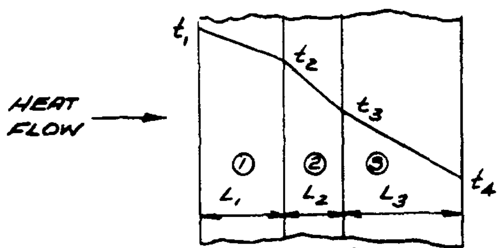


Fig. 6-a

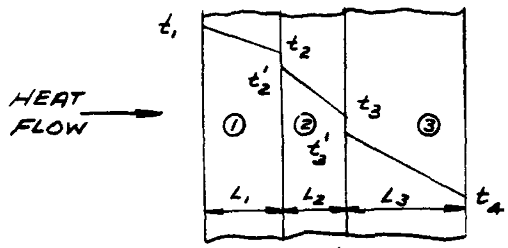


Fig. 6-b

Conduction through Composite Walls

Fig. 6-a shows a composite wall or bar insulated on the ends and made up of three different materials. It is assumed that the materials make perfect contact at the joints, a condition which is difficult to attain unless the materials are metals and bonded to each other by solder or other such means. Perfect contact at the joints eliminates the high resistance to heat

transfer caused by any surface roughness with accompanying air films between the joints. The equation for the heat flow is:

$$q = \frac{A\,(t_1 - t_4)}{\dfrac{L_1}{k_1} + \dfrac{L_2}{k_2} + \dfrac{L_3}{k_3}} \tag{14}$$

Each term in the denominator is a resistance to heat transfer and, since this is a series thermal circuit, the resistances are additive. The temperature gradient across each material is proportional to its thermal resistance. For example, the temperature drop for the second material is:

$$t_2 - t_3 = \frac{\dfrac{L_2}{k_2}}{\dfrac{L_1}{R_1} + \dfrac{L_2}{R_2} + \dfrac{L_3}{R_3}}\,(t_1 - t_4) \tag{15}$$

In cases where the materials are not actually bonded together, the thermal resistance at each joint should be considered. Fig. 6-b shows a composite wall or bar of three materials with imperfect thermal joints, having resistance at each joint. Since there is an abrupt temperature drop at each joint, there are five thermal resistances: one for each material and one for each contact or joint. The equation for the heat flow is:

$$q = \frac{A(t_1 - t_4)}{\dfrac{L_1}{k_1} + R_{1-2} + \dfrac{L_2}{k_2} + R_{2-3} + \dfrac{L_3}{k_3}} \tag{16}$$

where:

R_{1-2} and R_{2-3} are the thermal contact resistances offered by the first and second joint respectively. This contact resistance is complex because of the nature of the variables affecting it, such as the surface finish or roughness, the flatness of the contacting surfaces, the pressure holding adjacent materials together, and the materials used. It appears reasonable to neglect contact resistance where two surfaces are welded, soldered, or brazed together so that the contact is practically perfect. On the other hand, if two surfaces are not so bonded, the thermal contact resistance should be estimated by reference to the available published information. This information (see references 1 and 2 in Bibliography Appendix D) presents the contact resistance in terms of an equivalent coefficient of heat transfer, $h_{contact}$ with units of watts/(sq.in.)(^{o}C) or Btu./(hr.)(sq.ft.)(^{o}F). Thus the contact resistance R is expressed by:

$$R = \frac{1}{h_{contact}} \tag{17}$$

e. Notes on Contact Resistance

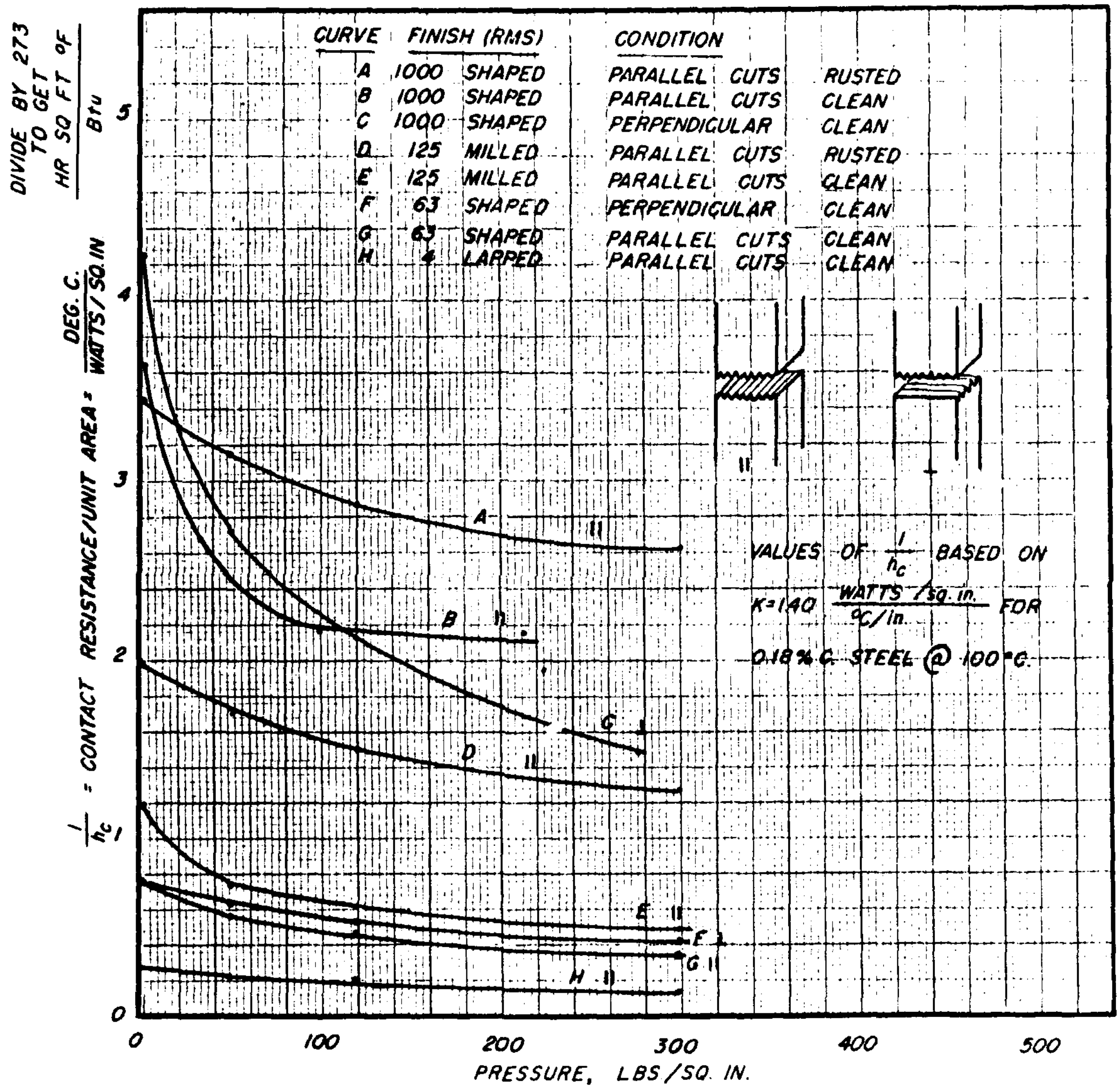
The contact resistance investigation results of reference (1) are reproduced in Fig. 7, in which contact resistance is correlated as a function of contact pressure using various steel surfaces. The surface roughness is indicated by the RMS index value which is the root-mean-square value of the heights and depths of the minute hills and valleys which form a machined surface. Thus, a lapped surface with an RMS of 4 (4 millionths of an inch) would constitute an extremely smooth surface.

Figure 7 shows a wide variation in contact resistance, especially at the lower contact pressures. It is of interest to convert the contact resistance into an equivalent length of material whose resistance due to pure conduction would be the same. For example, a contact resistance of 1.0^{o}C/watts/sq.in. is equivalent to 1.0/1.18 = 0.85 additional inches of steel in pure conduction. The 1.18 figure is the k value for mild steel in watts/(sq.in.)(^{o}C)/in. Hence, a relatively rough surface contact may easily result in a higher thermal resistance than the metal itself. Reference (1) indicated that aluminum foil placed in the joint decreases the thermal resistance.

Reference (2) lists considerable experimental data for joints of various materials. Two levels of roughness were used; 10 RMS was considered as smooth, and 50 to 100 as rough. The surfaces were clean and flat to ± 0.0001 in. Figures 8 and 9 correlate contact resistance for these different joints with pressure. Note that the contact resistance of the aluminum joint decreases more rapidly with pressure than for the steel joint. Table (2) summarizes the contact resistance of the various joints, all tested at 10 psi. The last column gives the resistance of the joints when filled with oil. These general conclusions follow:

(1) The thermal resistance of dry joints decreases linearly with pressure for steel. The thermal resistance of dry joints decreases exponentially with pressure for bronze and aluminum.

(2) The thermal resistance of both dry and oil-filled joints decreases with a decrease in roughness.



CONTACT RESISTANCE AS A FUNCTION OF CONTACT PRESSURE

Fig. 7

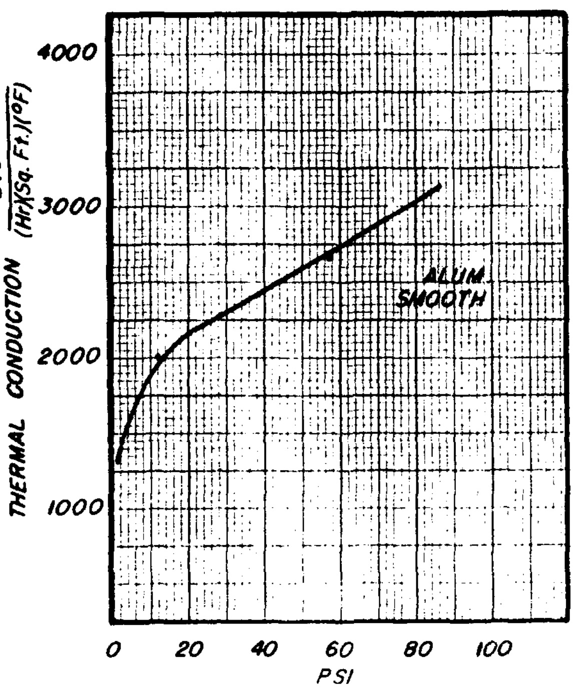


Fig. 8

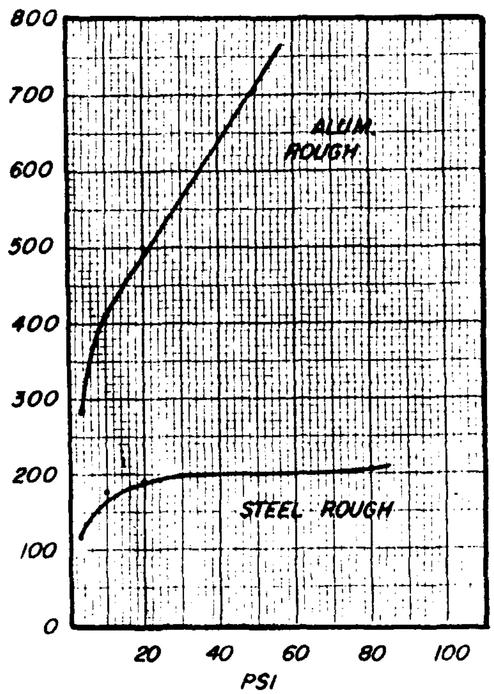


Fig. 9

THERMAL CONDUCTIVITY AS A FUNCTION OF CONTACT PRESSURE

TABLE 2

Comparison of Thermal Conductance Measurements at 10 psi

Joints		Surface Roughness RMS, Microinches Surface #1	Surface #2	Mean	Thermal Conductance BTU/(hr)(sq.ft)($^{\circ}$F) 300°F Dry	500°F Dry	300°F Oil
Steel	Smooth vs. Smooth	3	3	3	2200	3600	-
	Rough vs. Rough	70	85	78	400	800	1350
No.1 Alum.*	Smooth vs.Smooth	16	17	16	1800	3500	-
	Rough vs. Rough	60	60	60	1300	1500	2000
No.2 Alum.**	Smooth vs.Smooth	15	10	13	1900	2500	-
	Rough vs. Rough	20	50	50	500	650	1600
Bronze	Rough vs. Rough	70	80	75	800	1200	1200
No.1 Alum.*	Smooth vs.Smooth Steel Surface	15	90	66	800	-	1600

Thermal Conductivity of oil, $k_x = 271$ to 398×10^{-6} cal./(cm)^2(sec.)(deg.C)/cm

* Alcoa No. A-51-S

** Alcoa No. 18-3

Also:

Steel is SAE 4140
Bronze is AMS 4846

(3) At a given temperature, pressure and roughness, the thermal resistance of both dry and oil-filled joints decreases in the order: steel, bronze and aluminum.

(4) The thermal resistance of dry joints decreases as the temperature increases. There is no such relationship for oil-filled joints.

(5) At 10 psi the thermal resistance of an oil-filled joint is about one-half that of a dry joint. The effect of the oil decreases at higher pressures.

(6) The thermal resistance decreases if one surface of a steel joint is copper plated.

<u>Example 2</u>: Conduction with and without contact resistance.

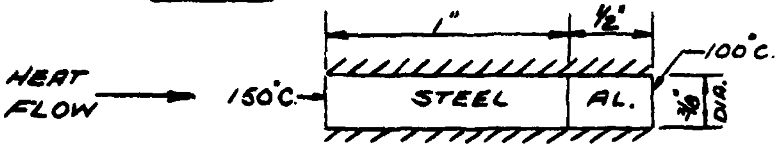


Fig. 10 Composite Bar with Contact Resistance

Figure 10 shows a composite bar, 3/8" diam., cross sectional area 0.110 sq. in., made up of 1" steel (k = 1.18) and 1/2" aluminum (k = 5.5) insulated on the sides. One end of the bar is maintained at 150°C and the other at 100°C. Assume a contact pressure at the dry joint of 10 psi and a contact resistance of 1.83 for contact surfaces of 70 RMS roughness. Problem: Determine the rate of heat transfer.

$$q = \frac{0.110\ (150 - 100)}{\frac{1.0}{1.18} + 1.83 + \frac{0.5}{5.5}} = \frac{5.5}{0.847 + 1.830 + 0.091} = \frac{5.5}{2.768} \qquad (16)$$

$$= 1.98 \text{ watts}$$

It is seen that in this particular case, the resistance to heat transfer caused by the joint constitutes $\frac{1.83}{2.768}$ x 100 or 66% of the entire resistance.

The heat flow rate, neglecting the contact resistance is:

$$q = \frac{0.11\ (150 - 100}{\frac{1.0}{1.18} + \frac{0.5}{5.5}} = 5.86 \text{ watts}$$

or almost three times as great as that wherein contact resistance is considered.

f. <u>Conduction Across Thin Air Gaps</u>

It has been found that heat transfer, other than that radiated, occurs between short vertical surfaces by gaseous conduction for distances up to about 1/4" (see reference 5). Beyond this distance a convective effect becomes apparent. At distances greater than 1/2" the convective heat transfer increases and approaches, for distances greater than 1", a value equivalent to that for a heated surface in free air having the temperature of the cooled surface.

In dealing with thin air gaps, say 1/4" or less, it can be assumed that the primary mode of heat transfer (other than radiation) is by conduction and the resistance can be estimated by:

$$R = \frac{L}{k}$$

where L is the thickness of the air gap and k is the thermal conductivity of air (see Table 18) at the average air temperature.

g. <u>Conduction Through Other Shapes</u>

The foregoing shows that pure conduction through a wall or bar, cylinder and sphere is relatively simple. All other shapes are complex and usually must be solved by graphical or numerical means. Where conduction through complicated shapes is accompanied by convection or radiation or both, it is difficult to define the problem mathematically. This results from the complexity of the thermal paths associated with densely packed electronic equipment. In certain instances these paths include "end effects" which are disregarded in conventional heat transfer design. Heat flows from regions of high to low temperature by conduction, radiation, convection and combinations of all these modes. Further, one mode affects the other so that densely packed electronic equipment represents a thermal circuit wherein it is difficult to apply an exact analytical treatment. However, there are instances which are readily applicable to an analytical treatment, an example of which is forced convection over electronic tubes wherein the heat transfer occurs primarily by only the one mode. Further, these complexities should not give the electronic engineer the impression that thermal design is impossible or even very difficult. The heat transfer characteristics of typical assemblies or subassemblies can be determined, if necessary, empirically, and thermal "bench marks" established.

h. Conduction Cooling Notes

Reference 10 presents the results of experimental investigation of cooling electronic parts by metallic conduction. One of the conclusions was that "due to the complex nature of heat transfer when pure conduction is taking place concurrently with other modes of heat transfer, an adequate general correlation of the data is not practical." It was found, however, that "components largely made of metal and fastened in firm metal to metal contact with a cooled surface, are cooled satisfactorily by this method." In general, the largest metallic surface area of the part should be in contact with the cooled surface. Further, it is stated that under good thermal contact, as much as 50% of the total heat dissipation may be by conduction. Also, "the temperature rise of components cooled by free convection in air at ground level pressure may be decreased 30 to 45% by fastening the component with metal-to-metal contact to a cooled surface at the same temperature as the ambient air, depending on the configuration and method of mounting the component."

In summary, the following recommendations are important regarding conduction cooling:

(1) Electronic parts, made largely of metal, may be efficiently cooled by conduction provided there is good metal-to-metal contact, preferably by soldering.

(2) The conduction path should be as short as possible, that is, the part should be so placed that its smallest dimension is perpendicular to the cooled surface. Also, if possible, the largest metallic surface of the part should be in contact with the cooled surface.

(3) Parts, other than metal ones, may be cooled by conduction, although less effectively. It is imperative that the thermal resistance at the contact plane be low, which can only be obtained if the surfaces are flat and smooth and if the two surfaces are in contact under pressure.

2. Free Convection in Gases

a. General

Heated bodies situated in a gas (or liquid) may lose an appreciable percentage of heat energy by free convection. Usually the gas is air, but other gases which exhibit higher coefficients of heat transfer, such as helium, are sometimes used. Heat transfer coefficients of gases in free convection are usually very low when compared with liquids and the thermal resistance due to the free convection film may be the greatest and, hence, the governing factor in a thermal circuit.

b. Units

A consistent set of units must be used in both free and forced convection calculations. It may be convenient to use the mechanical engineering system of units since the properties of fluids are usually listed in this system. Table 3 lists the nomenclature and the engineering system of units which may be used in both free and forced convection equations.

TABLE 3

Engineering System of Units
Free and Forced Convection Equations

Symbol	Nomenclature	Units
h_c	Convection coefficient of heat transfer	Btu/(hr.)(sq.ft.)(^{o}F)
L	Characteristic length	ft.
k	Thermal conductivity	Btu/(hr.)(ft.)(^{o}F)/ft.
g	Acceleration due to gravity	4.17×10^{8} ft./hr.^{2}
β*	Coefficient of thermal expansion	Cu.ft./(cu.ft.)(^{o}F)
ρ	Density	Lbs./cu.ft.
μ	Viscosity	Lbs./(ft.)(hr.)
c	Specific heat at constant pressure	Btu/(lb.)(^{o}F)
V	Velocity	Ft./(hr.)

* For a gas, the coefficient of thermal expansion is numerically equal to the reciprocal of the absolute temperature, $1/{}^{o}R$.

c. Theoretical Considerations

The basic equation for the free convection film coefficient for any fluid, either liquid or gaseous, is:

$$\frac{h_c L}{k} = C \left(\frac{g \beta \Delta t L^3 \rho^2}{\mu^2}\right)^m \left(\frac{c\mu}{k}\right)^n \tag{18}$$

The symbols and units are given in Table 3. Δt is the temperature difference between the surface and the fluid. C is a constant which, in general, depends on the shape of the surface, while the exponents m and n depend on the magnitude of the groups of variables in the parentheses.

The term in the left of equation (18), $h_c L/k$, is a dimensionless group called the Nusselt number, Nu. The group of terms in the first set of parentheses on the right is also dimensionless and is known as the Grashoff number, Gr., while the terms in the second parentheses are a dimensionless group known as the Prandtl number, Pr. These three groups are important in free convective heat transfer, and equation (18) has been used to correlate experimental data. Furthermore, this equation can be derived by theoretical considerations. It has been found experimentally that the magnitude of m and n are very nearly equal and equation (18) may be rewritten in the form

$$h_c = C \frac{k}{L} (a L^3 \Delta t)^m \quad (19)$$

where "a" is defined as:

$$a = \frac{g\beta \rho^2 c}{\mu k} \quad (20)$$

The significant dimension L is a function of the character and position of the convecting surface. The following table lists L values for the more common shapes.

TABLE 4

SIGNIFICANT DIMENSION "L"

Surface	Position	Length
Plane	horizontal	$\frac{(\text{Length})\times(\text{Width})}{\text{Length} + \text{Width}}$
Plane (rectangular)	vertical	vertical height but limited to 2 ft.
Plane (non-rectangular)	vertical	$\frac{\text{area}}{\text{horizontal width}}$
Plane (circular)	vertical	0.785 x diameter
Cylinder	horizontal	diameter
Cylinder	vertical	height of cylinder but limited to 2 ft.
Sphere		0.50 x diameter

For irregularly shaped electronic parts, the L value would be that of the most similar shape or surface in the foregoing table.

The exponent "m" is dependent on the value of $aL^3\Delta t$. In the range from $aL^3\Delta t = 10^3$ to 10^9 its value is about 0.25. If the magnitude of $aL^3\Delta t$ exceeds 10^9, the value of m increases to 0.33. However, the range 10^3 to 10^9 includes almost all electronic free convection calculations.

Thus, the free convection equation can be written:

$$h_c = C \frac{k}{L} (aL^3 \Delta t)^{0.25} \qquad (21)$$

Reference (11) lists values of the constant C for various shapes and surfaces. In general, these values are for relatively large dimensions when compared with subminiature electronic parts. Examples of applicable shapes are boxes used to house electronic parts. The values are listed in Table 5.

TABLE 5

Values of C to be Used in Equation (21)

Shape and Position	C
Vertical plates	0.55
Horizontal cylinders (pipes and wires)	0.45
Long vertical cylinders	0.45-0.55
Horizontal plates facing upward	0.71
Horizontal plates facing downward	0.35
Spheres (L = radius)	0.63

In using equation (21) it is convenient to use the Btu-°F-foot system. For a gas, the coefficient of thermal expansion is equal to the reciprocal of the absolute temperature or 1/°R where °R = °F + 460. The properties of air at sea level pressure, including the "a" term, are given in Table 13 (Appendix C).

For free convection in air, reference (6) includes a chart which permits solution of the foregoing equation with a very minimum of calculation. This chart has been reproduced in Figure 11. It is only for air and cannot be used for other gases. While the usual gas is air there are instances wherein other gases such as helium might be used within a sealed unit to advantage. In this case calculations must be made using equation (21).

d. Example No. 3

The following example illustrates the use of equation (21) and also the use of the free convection chart.(in Appendix envelope).

(1) Problem

A metal container, 24" long x 12" wide x 12" high, situated in free air at 35°C, has a surface temperature (average) of 85°C. How many watts can be dissipated (exclusive of radiation and conduction) by free convection from the top surface only to the air? It is assumed that the space surrounding the container will remain at 35°C (well ventilated)

(2) Solution by Calculation

Step (1) - Determine film coefficient by use of equation (21).

$$h_c = C \frac{k}{L} (aL^3 \Delta t)^{0.25}$$

$$t_s = 85^\circ C = 185^\circ F$$

$$t_a = 35^\circ C = 95^\circ F$$

Average temperature of air film = $\frac{185 + 95}{2} = 140^\circ F$

Pertinent properties of air at 140°F (from Table 18)

$$k = 0.0168 \text{ Btu/(hr.)(ft.)(}^\circ\text{F)}$$

$$a = 0.896 \times 10^6$$

$$C = 0.71 \text{ (for horizontal plate facing upward)}$$

$$\Delta t = 185 - 95 = 90^\circ F$$

$$L = \frac{12 \times 24}{12 + 24} = 8 \text{ in.} = 0.667 \text{ ft.}$$

$$h_c = 0.71 \frac{0.0168}{0.667} (0.896 \times 10^6 \times 0.667^3 \times 90)^{.25}$$

$$= 1.25 \text{ Btu./(hr.)(sq.ft.)(}^\circ\text{F)}$$

Step (2) - Determine total heat transfer rate

$$q = h_c A \Delta t \text{ (Basic equation)}$$

$$A = 12 \times 24 = 288 \text{ sq. in.} = 2.0 \text{ sq. ft.}$$

$$q = 1.25 \times 2.0 \times 90 = 225 \text{ Btu/hr.}$$

Step (3) - Convert to watts

$$225 \text{ Btu./hr.} \times \frac{1}{3.415} = 66.0 \text{ watts total heat dissipation by free convection from top surface only.}$$

Note: The calculations for the heat dissipation from the sides and bottom of the box may be made in a similar manner using appropriate constants from Table 5.

(3) Solution by Chart, Fig. 11

Enter top right quadrant at 85°C surface temperature and proceed horizontally to 35°C air temperature line. Follow vertical line downward to 30 in. Hg. pressure (atmospheric) in lower right quadrant. Proceed horizontally to horizontal plane face-up line in lower left quadrant. Proceed vertically upwards to significant dimension line of 8.0 in. in upper left quadrant (use upper group of lines). Proceed horizontally to right and read 0.227 watts per sq. in.

Total watts from upper surface is:

$$0.227 \times 12 \times 24 = 65.4 \text{ watts}$$

(this is in close agreement with the calculation method)

Note: A similar example wherever the heat dissipation is given and the temperatures are to be found is presented on page 96.

e. Free Convection from Small Confined Parts

Ref. 5 cites free convection tests of tubes, resistors, relays and transformers in confined spaces. Due to enclosure effects and the irregular configuration of the parts together with their smallness, it was found that the convection heat transfer coefficients were greater than those calculated by using the constants in Table 5. The test data are best fitted by the following equation:

$$Nu = 1.45 \, (Gr \times Pr)^{0.23} \tag{22}$$

(where Nu, Gr, and Pr are the Nusselt, Grashof and Prandtl numbers respectively).

For the miniature and subminiature tubes tested, the height was used as the significant dimension L. For the horizontal resistors, L was defined as the reciprocal of the sum of the reciprocals of the diameter and length. For the relays and transformers, L was taken as the vertical height.

Since the total heat transfer rate q by free convection from the surface area A is given by:

$$q = hA\Delta t \qquad (23)$$

equations (20) and (21) may be combined and solved for the difference between the surface and fluid temperature. Ref. 10 changes the exponent of the Grashof and Prandtl numbers from 0.23 to 0.25 (for convenience) which results in:

$$\Delta t = 0.75 \frac{L}{a}^{0.2} \frac{q}{Ak}^{0.8} \qquad (24)$$

with all units in the Btu.-foot-degree F system.

The application of equation (22) is given in an example in the Liquid Cooling Section. These free convection equations are applicable to any fluid, liquid or gaseous.

3. Radiation

a. General

The basic equation for the net exchange of radiant energy from a surface at an absolute temperature of T_1 to a surface at a lower temperature, T_2, is

$$q_r = \sigma A\ F_e\ F_a\left[(T_1)^4 - (T_2)^4\right] \qquad (25)$$

where:

σ is the Stefhan-Boltzmann constant

A is the area of the higher temperature surface (in most cases)

F_e is an emissivity factor

F_a is a configuration factor

The units to be used in the general radiation equation in the two systems are:

	Btu-foot-hour-$^\circ$F System	Watt-Inch-Sec.$^\circ$C System
q_r	Btu./hr.	Watts
σ	0.173×10^{-8} Btu./(hr.)(sq.ft.)($^\circ$R)^4	0.0037×10^{-8} watts/(sq.in)($^\circ$K)^4
A	sq. ft.	sq. in.
F_e	Same in both systems - dimensionless	
F_a	Same in both systems - dimensionless	
T	$(T)^\circ R = 460 + (t)^\circ F$	$(T)^\circ K = 273 + (t)^\circ C$

The emissivity factor F_e allows for the departure of two radiating surfaces from ideal blackness or unity emissivity. In general, F_e is a function not only of the emissivities of the two radiating surfaces but of their geometric arrangement as well. For parallel planes which are large compared to their distance apart and also for a completely enclosed body which is large compared to the enclosing body, F_e is given by:

$$F_e = \frac{1}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \tag{26}$$

Where ϵ_1, and ϵ_2 are the emissivities of the two surfaces. For a completely enclosed body which is small compared to the enclosing body, such as an electronic box in a large compartment or room, the emissivity of the enclosing surface has little effect on F_e and

$$F_e = \epsilon_1 \tag{27}$$

where ϵ_1 is the emissivity of the enclosed body. These are the more general configurations, but for others, reference should be made to "Introduction to Heat Transfer" by Brown and Marco (ref. 11).

Tables 20 and 21 list emissivity values of various surfaces. Dull, dark surfaces are good absorbers (or emitters) and have high emissivity values. Polished surfaces have low values and can be used as radiation shields to protect parts from radiant heat sources.

In the case of enclosed bodies, the area A in the radiation equation is that of the enclosed body. The configuration factor F_a takes into account the geometry of the radiating surfaces and the fact that not all of the radiation from one surface may reach the receiving surface. In most cases, such as the large parallel planes and enclosed bodies, F_a is unity. For certain other configurations F_a may vary widely and much information is given in reference 11.

Equation (25) may be written in a slightly different form for ease of solution, Using the Btu-hr.-sq.ft.-°R system, it is:

$$q_r = 0.173 \ A \ F_e F_a \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right] \tag{28}$$

and in the watt-sec.-sq.in.-°K system:

$$q_r = 0.0037 \ A \ F_e \ F_a \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right] \tag{29a}$$

b. <u>Example (4) of Use of Radiation Equation</u>:

<u>Problem</u>: An electronic assembly is housed in a steel box painted with a dull paint. Its dimensions are 6" x 12" x 6" high. It is mounted in a rack such that the total surface area is exposed. The average surface temperature is 150°C and the objects surrounding the box are at 80°C. Also, the surfaces surrounding the box are in relatively close proximity. Estimate the radiant energy dissipated from the box.

<u>Solution</u>:

A_1 = 360 sq. in.

F_e = It is assumed that the emissivity of the surrounding objects is 0.90. From Table 20 the emissivity of the surface of the box is 0.94. Since the surrounding objects are in close proximity to the box, F_e is calculated by

$$F_e = \frac{1}{\frac{1}{0.94} + \frac{1}{0.90} - 1} = 0.852$$

F_a = 1.0

$T_1 = 150 + 273 = 423^\circ K$

$T_2 = 80 + 273 = 353^\circ K$

From equation (29)

$$q_r = 0.0037(360)(0.852)(1.0)\left[\left(\frac{423}{100}\right)^4 - \left(\frac{353}{100}\right)^4\right]$$

q_r = 186.8 watts total radiant heat dissipation.

An alternative solution is by use of the two radiation charts Fig. 12, from ref. (6), one for the lower range and the other for the higher range of temperature. Using the lower range chart, 150°C is located at the left and one proceeds horizontally to the right to the receiving temperature of 80°C and then vertically downward to the horizontal scale reading 0.615 watts/sq.in. This is for the ideal black body only and must be multiplied by F_e and F_a as well as the total area. Thus,

$$q_r = 0.615 \times F_e \times F_a \times A$$

$$= 0.615\,(0.852)(1.0)(360) = 188.8 \text{ watts}$$

Conversely, if the heat dissipation of the above box is known, the surface temperature can be computed:

$$q_r = 185 \text{ watts}$$

$$185 = 0.0037\,(3.60)(.852)(1.0)\left[\left(\frac{T}{100}\right)^4 - \left(\frac{353}{100}\right)^4\right]$$

$$\left(\frac{T}{100}\right)^4 = \frac{185}{1.035} + 164$$

$$\frac{T}{100} = \sqrt[4]{342.5} = 4.262$$

$$T = 426^{\circ}K$$

$$T_1 = 426-273 = 153^{\circ}C$$

c.. Comparison of Radiation with Free Convection

To show that radiation is an important mode of heat transfer, it is of interest to compare it with free convection. In the foregoing problem, free convection acts simultaneously with radiation. Assume that the surrounding air is at 80°C. The following tabulated calculations for the free convective heat transfer are from the free convection chart, Fig. 11.

Example (5)

FREE CONVECTION FROM BOX OF EXAMPLE (4)

Surface	Significant Dimensions, in.	Area Sq. in.		Watts/ sq. in.		Total Watts
Top	$\frac{12 \times 6}{12 + 6} = 4"$	72	x	0.393	=	28.3
Bottom	$\frac{12 \times 6}{12 + 6} = 4"$	72	x	0.193	=	13.9
Sides	height = 6"	216	x	0.276	=	59.7
				Total	=	101.9

Hence, in this case, the total dissipation from the box is 188.8 (radiation) plus 101.9 (convection), or 290.7 watts, of which 65 percent is by radiation.

Radiation and convection usually occur simultaneously. Since free or forced convection is usually described by a convective coefficient of heat transfer h_c, it is convenient to use an equivalent coefficient of radiation, h_r, so that the two coefficients are additive. Thus, for a surface at temperature t_s and of area A transferring heat by convection to the surrounding air at temperature t_a and simultaneously transferring heat by radiation to radiant receiver surfaces at temperature t_r, the total heat transfer rate from the surface by these two modes is:

$$q_t = q_c + q_r = (h_c + h_r) A \ (t_c - t_a) \qquad (29b)$$

The radiation coefficient is:

$$h_r = \frac{\sigma F_e F_a (T_s^4 - T_r^4)}{t_s - t_a} \qquad (29c)$$

If the surrounding receiver surface temperature is equal to the air temperature, T_a is substituted for T_r.

d. Design Notes on Radiation

Even though the thermal circuits in densely packaged electronic equipment are complex, approximate radiation calculations may be made. There are several design principles which may be used to advantage and which should be kept in mind:

(1) The heat transferred by radiation may exceed that transferred by convection. Thus, radiation is an important mode of heat transfer.

(2) For maximum heat transfer by radiation, "black" surfaces must be used. This should not be interpreted to mean that all surfaces should be painted black. Some judgement must be exercized. For example, vacuum tubes should not be painted black because the emissivity of glass at temperatures as low as 100°C is equal to that of the best black paint and with increasing temperatures becomes slightly greater than that of the paint.

(3) For a given difference between radiating and receiving surface temperatures, the higher the level of temperature, the greater will be the radiant heat dissipation. This is due to the heat transfer being proportional to the difference in the fourth powers of the absolute temperatures. Thus, theoretically, parts should be operated at their maximum temperature ratings to realize maximum heat transfer by radiation.

(4) Uncontrolled radiation can cause impaired reliability. It is desirable to protect temperature sensitive, or low rated temperature parts, from overheating, due to their

proximity to higher temperature heat sources. Hence, low temperature parts must be located so that they do not "see" these sources or radiation shields must be used. Thin, highly polished sheet metal shields placed between such parts can be very effective as radiant heat barriers. The shield should be polished on both sides. Also, it is desirable that the shield be soldered to the case or chassis to provide a good conductive heat path.

(5) Placement of parts in an electronic assembly to provide maximum radiant heat dissipation requires careful consideration. For example, a tube surrounded by other tubes could dissipate little radiant heat with a consequent higher temperature rise than its neighbor. There may be occasions due to assembly limitations wherein a certain part must be given special treatment to provide adequate cooling, one example being a tightly fitting tube shield soldered to the case or chassis to provide a highly conductive thermal path away from the tube.

4. Radiation and Gaseous Conduction

Heat transfer can occur in gases primarily by conduction, across small gaps between surfaces at different temperatures, if convective effects are suppressed. Radiation and gaseous conduction were studied in Ref. (10) with respect to the cooling of vacuum tubes and relays. Each part to be studied was enclosed by a cylindrical brass baffle blackened on the inside and immersed in a constant temperature bath. The spacing between the tubes and baffles was 0.20 inch.

It was concluded that exact mathematical correlation of the data was difficult, since the heat was dissipated simultaneously by both radiation (a function of the difference of the fourth power of absolute surface temperatures) and gaseous conduction (proportional to the first power of the absolute temperatures). Further, nonisothermal surfaces and effective surface areas of irregularly shaped parts complicated the analysis. However, the tests did produce important general conclusions which are in accordance with heat transfer theory. These are listed as follows:

a. The heat transfer between confined, short, vertical surfaces occurs only by gaseous conduction and radiation for distances up to about 0.25 inch. Convection was only apparent with a greater separation. Thus, in densely packaged equipment, free convection effects are probably almost absent.

b. Heat transfer by gaseous conduction is inversely proportional to the distance or spacing. Hence, for optimum heat transfer the spacing must be very small, in which case the heat transfer by gaseous conduction may exceed that by normal convection of unconfined parts.

c. For cylindrical parts surrounded by closely spaced blackened shields, the heat transfer rate may be estimated by the following two equations:

Conduction:

$$q_c = \frac{2.729\, k_{air}\,(t_p - t_c)}{\log_{10}\left(\frac{D_e}{D_p}\right)} \qquad (30)$$

where:

k is the thermal conductivity of the air or gas within the space

t_p is the estimated mean temperature of the part

t_s is the temperature of the shield

D_s and D_p are the diameters of the shield and part respectively

Radiation:

$$q_r = 0.173\, F_e F_a\, A_p \left(\frac{T_p}{100}\right)^4 - \left(\frac{T_s}{100}\right)^4 \qquad (31)$$

where:

F_e is the emissivity factor having a value between 0.85 and 0.95 for most electronic parts including electronic tubes, as long as any metal parts are not bright or polished

F_a is the configuration factor, being unity for the configuration of a cylinder enclosing a part

A_p is the area of the part

T_p and T_s are the absolute temperatures of the part and shield surface respectively

The total heat transfer rate is the sum of q_c and q_r

Problems involving combined modes of heat transfer are solved by trial and error methods.

d. The tests of reference (10) showed, as could be predicted from heat transfer theory, that increasing the tube and shield temperatures the same amount results in greater heat transfer by radiation while the heat transfer by gaseous conduction remains the same. Thus, for the same heat dissipation, the temperature difference between part and shield temperature is less at an elevated temperature level than at a low temperature level. This is due to the increased effectiveness of radiant heat transfer in accordance with the fourth powers of the temperatures.

C. NATURAL METHODS OF COOLING ELECTRONIC PARTS

1. General

Most electronic parts have been designed for cooling in free air at one atmosphere by natural means, primarily by radiation and free convection. Conduction cooling is usually the less significant mode. The ideal free air environment for electronic parts is seldom obtained in conventional electronic equipment, and probably is never achieved in miniaturized equipment. This means that the parts must either be derated or provided with supplementary cooling. Such cooling techniques by natural means are discussed in this section. The recommended method may not always be compatible with a particular design and, where possible, alternate means are presented.

2. Vacuum Tubes

a. Hot Spot Locations

The modes of heat transfer within vacuum tubes and the location of the hot spots produced during operation in free air are discussed in Section X. Note that the primary hot spot is on the envelope opposite the plate, that the secondary hot spot is at the base near the leads, that tubes should be cooled in a manner that will reduce the thermal gradients in the glass and that tubes must be cooled primarily by removing heat from the glass envelope (See Section X). In general, the hot-spot temperature rise on the envelope opposite the plate of a tube in free air will be of the order of twenty per cent greater than the average envelope temperature rise.

b. Shields for Removing Heat from Tube Envelopes

(1) General

This section is concerned with the ability of tube shields to remove heat from the tubes which they enclose.

A tube shield should also act as an electrical shield around the tube to reduce interaction due to stray fields. It should support the tube securely in its socket against vibration and impact in any plane, and it should protect the tube and its leads from mechanical injury.

There are three heat transfer paths from a bare tube: radiation from the envelope to surfaces which the envelope "sees", either natural convection or gaseous conduction from the envelope to the environmental air or gas surrounding the tube, and conduction along the tube lead wires. Since the greatest concern is the glass envelope, this discussion will stress the former mode. In brief, heavy gage, short length wire leads which are thermally grounded, tend to increase heat transfer by conduction. The terminals of the leads (away from the tube) must be kept cool if conduction is to be at all appreciable. In general, conduction along and convection from long lead wires on subminiature tubes, is not appreciable.

The major mode of heat transfer from a bare vacuum tube is radiation. This is because the heat transfer by radiation is a function of the difference of the fourth power of the absolute temperatures, whereas, convection and gaseous conduction are functions of only the first power of temperature difference. If a tube is surrounded by lower temperature surfaces which are at distances greater than one inch from the tube, natural convection will occur. The heat transferred to these surfaces by convection will be less than that transferred by radiation. If the surrounding surfaces are very close, say, less than one-half inch away, and if the tube is enclosed in an airtight container, then free convection becomes ineffective and heat will be transferred by gaseous conduction to a smaller degree than that by radiation.

In order to be effective in removing heat from the envelope, the first and most important consideration is to provide a minimum of contact resistance between the tube shield, its base and the chassis or mounting surface. The mounting surface should be of metal. Nothing is gained by mounting tube shields on, for example, a phenolic chassis, or on materials of low thermal conductivity. Such materials act as thermal insulators and it is easily possible to overheat a well shielded vacuum tube, even when it is operated well within its dissipation ratings. Ideally, for maximum heat transfer, the tube shield should be soldered, brazed or bonded to the metal surface to obtain a near perfect contact. Riveting or bolting a shield to a metal surface leaves a thin air gap which constitutes an extremely high thermal resistance, probably much more than the resistance of the shield itself. For effective heat removal, this gap must be minimized and preferably eliminated. A poor surface contact may cause a shielded tube to operate hotter than a bare tube, even though other thermal considerations are incorporated in the design. It is not advisable to use a tube shield which is not thermally bonded to a cooler metal surface.

In addition, the shield should fit the tube envelope as tightly as possible to reduce the air gap to a minimum. Perfect contact with the bulb glass is difficult. One method which has found some use is to apply silicone grease between the tube and the shield. Unfortunately, this method is not usually suited to the maintenance techniques of the Armed Services. In general, the most practical method is to provide some flexibility in the shield to accommodate expansion and variation in bulb dimensions. For example, this can be accomplished by slotting or splitting the shield.

Further, it is advisable to increase the absorptivity of the inner surface of the shield to increase the heat transfer by radiation from the envelope. A brightly polished surface is a poor absorber and should not be used. A dull, oxidized and blackened surface is preferred. If heat transfer by radiation to the surroundings is desired, then the same dull surface should be used on the outer surface of the shield. On the other hand, if temperature sensitive parts and other tubes constitute the surrounding objects (in which case radiant heat transfer to these parts is not desired) then the shield's outer surface should be highly polished. Thus, the surrounding objects influence the design of the shield. In general, it is not recommended that radiant energy be deliberately expended inside an electronic case. The heat is radiated from the source and dispersed to other parts in an uncontrolled fashion.

(2) Miniature Tube Shields

(a) Conventional Shields

The standard type miniature tube shield, Fig. 13, is not too satisfactory from the heat transfer standpoint, especially at high heat concentrations. Such a shield is usually a heat barrier. The blanket of air enclosed between the shield and tube envelope is too thin for free convection currents to form and heat transfer from the tube to the shield is only possible by gaseous conduction

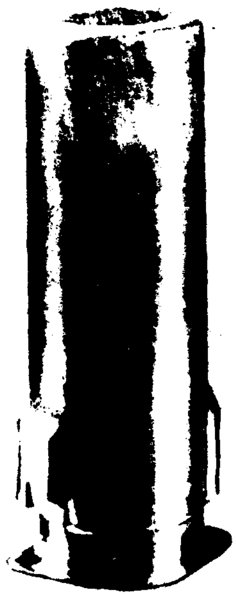


Fig. 13

JAN TUBE SHIELD

and radiation. Due to the low emissivity of the brightly finished shield, a large portion of the radiation is reflected rather than absorbed. Further, the average shield has a poor thermal contact with the chassis which prohibits heat conduction into the chassis.

Miniature tubes enclosed in standard nickel plated JAN type shields at room ambient temperature and pressure, show approximately a 45 per cent increase in hot-spot temperature rise over an unshielded tube. One organization has found that the provision of circumferential slots near the base of the shield to improve convection was ineffective (ref. 5). However, convection cooling was increased by cutting large vertical slots in the shield. With both vertical slots and a blackened standard JAN shield, the envelope hot-spot temperature seemed $8^{\circ}C$ cooler than the unshielded tube. However, this apparent temperature reduction was due to a shift in the hot-spot location caused by the convective cooling of the tube envelope. In general, an unshielded miniature tube will be cooler than a shielded tube. Therefore, in applications where electrostatic shielding is not required, a spring type "hold down" clamp will usually provide better cooling than a standard miniature shield, provided the radiation from the tube can be tolerated.

To summarize, the bulb temperature rise is least without any shield. Blackening a conventional miniature tube shield appears to be the only practical method of lowering the temperature rise, if such a shield must be used. Even so, the improvement is rather insignificant. The relatively poor ability of the conventional shield to facilitate heat removal from the tube is due, mostly, to the poor thermal attachment of the shield to the chassis. Conduction from the shield to the chassis is greatly impaired due to the method of attachment. *

(b) Radiation-Conduction and Radiation-Conduction-Convection Shields

With the radiation-conduction shield (see Fig. 14), heat transfer takes place by combined radiation and gaseous conduction from the tube to the shield and then, primarily, by metallic conduction from the shield to the cooled chassis. In such instances the thicker shields produce relatively lower temperatures, due to the larger heat conduction path. However, it has been found that increasing thickness results in diminishing returns. An increase of from 1/32 inch to 1/16 inch decreases the tube-to-sink temperature difference by 10 per cent, whereas, an increase from 1/32 inch to 3/16

* This work is continuing and will be reported in a future publication.

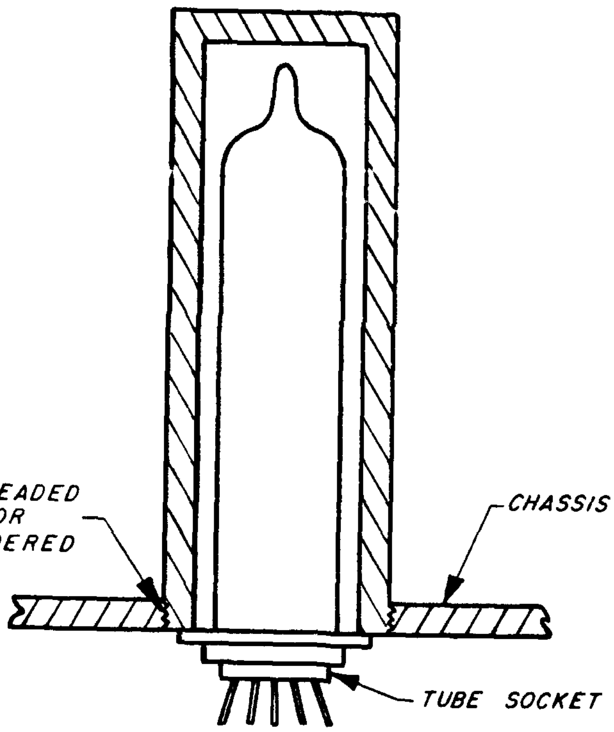


Fig. 14
CONDUCTION TUBE SHIELD

inch decreased the difference by only 20 per cent, (ref. 10).

The heat transfer rate from the radiation-conduction-convection shield is influenced by the size of the air gap as well as the number of slots. With small air gaps there is little free convection, and slotting the shield produces high temperatures. Slotting the plain shield increases the heat transfer only if the air gap is relatively large, say, 1/4 inch. The number of slots also affected the heat transfer although no correlation was apparent (See Fig.15).

Since a standard noval tube shield will just chamber the standard shield base made for a seven-pin miniature tube, it is possible to obtain a more effective convection cooled shield for use with nine-pin noval tubes by inserting the base of a seven-pin tube shield within a noval shield (see Fig. 16). The junction between the insert and the main tube shield is made by soldering, care being taken not to fill up the air space. This modification will permit convection between the tube envelope and the shield to reduce the envelope temperature. It is particularly applicable to tubes which are operated at full ratings in equipment of conventional construction.

(3) Subminiature Tube Shields

(a) General

The majority of subminiature tubes are used in radio frequency and voltage amplifier applications wherein about two watts per tube are dissipated. At this power level a single unshielded tube operating in free air at 25°C can attain an envelope temperature of 110°C at thermal equilibrium. Approximately 40% of the heat is removed by radiation while the greatest part of the remaining 60% of the heat is removed by free convection. When this same tube is placed in a subassembly with other similar tubes, heating by mutual radiation and convection is increased, and the tube envelope temperature can rise to 225°C at equilibrium. It is possible to alleviate this condition by using tube shields designed to remove heat from the tube envelope and transfer it into the chassis by metallic conduction.

An assortment of subminiature tube shields designed for conduction cooling have recently become available. A rough evaluation of several types of subminiature tube shields now in common use, along with some specially designed shields developed in this Laboratory, was conducted.* There are four general types of subminiature

* This work is continuing and will be reported in a future publication.

Fig. 15

RADIATION – CONDUCTION – CONVECTION SHIELD

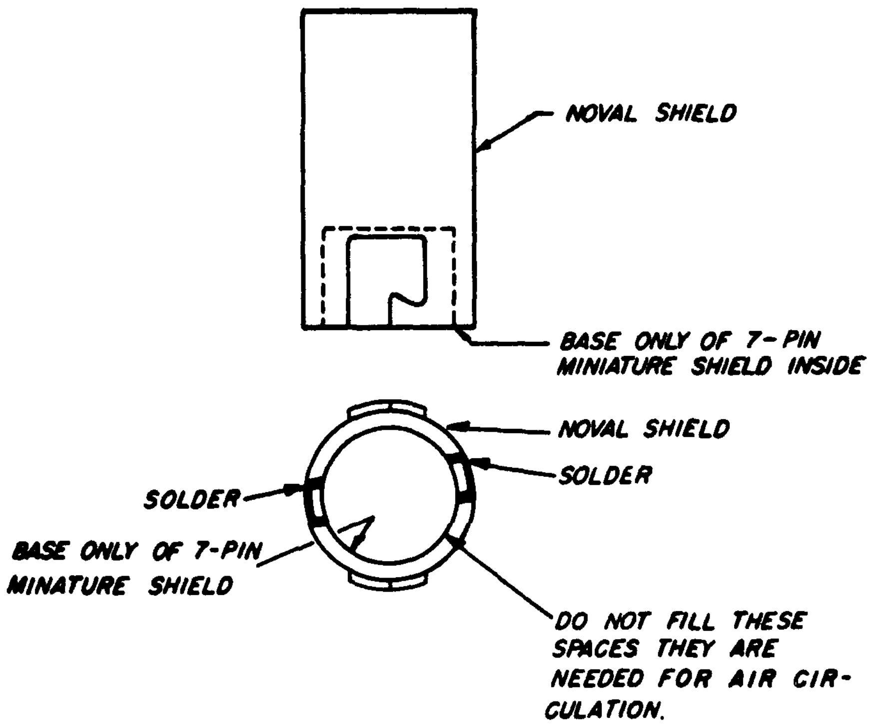


Fig. 16

IVES' SHIELD

tube shields: the wrap-around shield, the fuse clip type shield, the machined aluminum shield and the solid metal block type shield. The selection of a given shield depends upon the particular requirements of the electronic sub-assembly involved.

(b) The wrap-around shield (Fig. 17) is the most commonly used in current electronic equipment. The shield consists of a cylinder of springy metal which is formed to fit and wrap around a subminiature tube envelope. Heat is transferred from the glass envelope of the tube by radiation, direct conduction, and gaseous conduction to the shield. The shield in turn forms a conduction path of low thermal resistance to the chassis.

(c) Another type of subminiature tube mounting utilizes a fuse clip type configuration (Fig. 18) in which a spring metal clip is used to hold the tube in position. The fuse clip is usually riveted to the chassis or support. Some fuse clip mountings are long and cover a good part of the tube surface, thus providing a good conductive heat path away from the tube envelope. Small fuse clips have also been used to hold the tube in a favorable position for free convection and radiation cooling.

(d) The cylindrical shield consists of an aluminum tube shield machined out of a slotted aluminum tube with threads on its base so that it may be screwed into a threaded chassis (see Fig. 19f).

(e) An alternate method of mounting subminiature tubes has been to insert the tubes in a drilled metal block, the holes (Fig. 20) being slightly larger in diameter than the outside diameter of the tubes. Metal tube blocks of this type have been built as an integral part of the outside of equipment cases. Various methods have been used to hold the tubes in the block. In one instance, silastic type rubber was used in the form of a ring around the top and the bottom of the tube envelope to form a shock mounting for the tube. With such a mounting, the primary heat transfer modes from the tube envelope are by radiation and gaseous conduction to the metal block which, in turn, provides an excellent metallic conduction path to the surface of the equipment of subassembly case. Another technique involves wrapping the tube with corrugated aluminum, silver or copper foil prior to insertion into the metal tube block. The air gaps are reduced and conduction is increased.

(f) Evaluation of Subminiature Tube Shields

Initial Considerations:

The analysis of the thermal characteristics of tightly fitting tube shields is complicated by temperature gradients

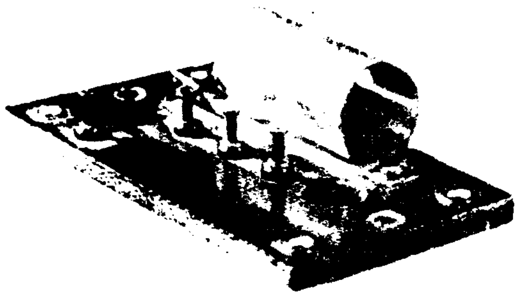


Fig. 17

WRAP-AROUND SUBMINIATURE TUBE SHIELD

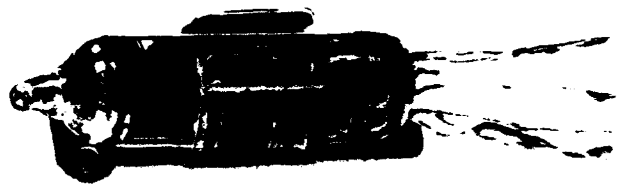


Fig. 18

FUSE CLIP TYPE SUBMINIATURE TUBE SHIELD

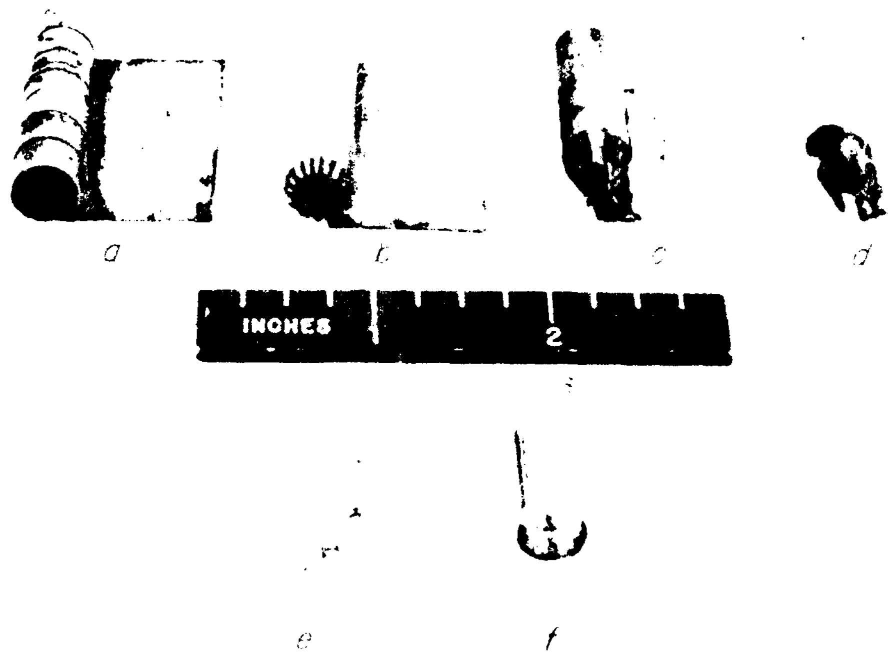


Fig. 19
SEVERAL TYPES OF SUBMINIATURE TUBE SHIELDS

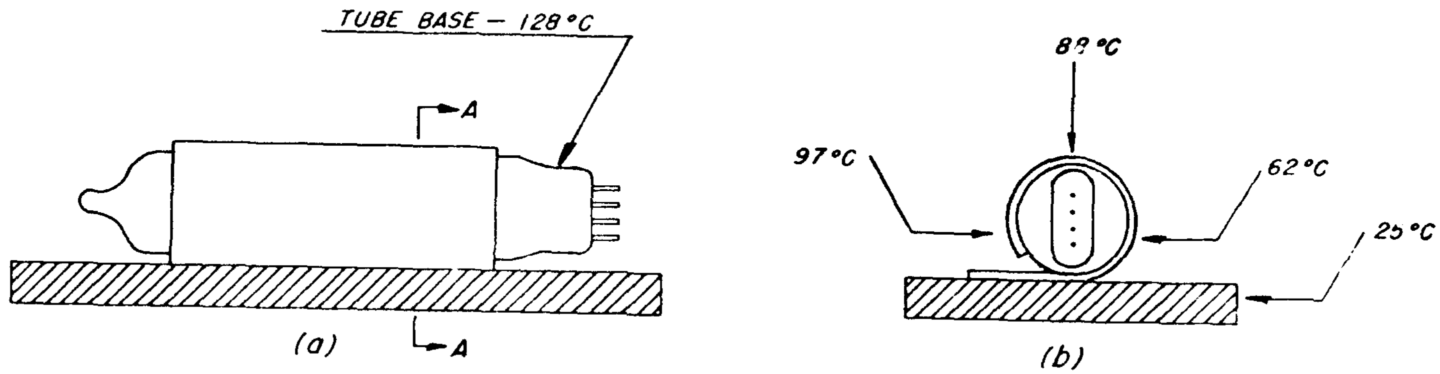
Fig. 20

DRILLED METAL BLOCK FOR MOUNTING SUBMINIATURE TUBES on the surface of the tube envelope, and the envelope surface temperature measurement errors due to the intimate contact between the envelope and the shield. It has been found that shield temperatures are misleading. In order to minimize those errors, small holes were drilled in the shields to be tested, and thermocouples were inserted and attached to the tube envelopes with Sauereisen cement. Some inherent error exists with this method since the thermocouples were on a part of the envelope which is cooled only by convection and radiation. In other instances, as an alternate, temperature indicating paints and lacquers were placed on the surface of the tube envelope. However, since the lacquer or paint was in contact with both the shield and the envelope, the indicated temperature was somewhere between that of the envelope and the shield. These techniques were considered satisfactory only for the determination of the relative merits of the shields tested. More accurate results could be obtained by inserting thermocouples into the envelope glass or by attaching thermocouples to several internal tube elements.

Each shield was mounted by soldering or by other low resistance means to a brass sink plate which was partially immersed in water. This provided an adequate heat sink whose temperature remained constant during these tests. Type CK5703 tubes were used throughout and operated at 3 watts input to thermal equilibrium. Extreme care was used to make sure that the entire electrical input to the tubes was dissipated (no electrical output). All tests were conducted at 24°C ambient and sink temperature at sea level pressure. Where possible, the tube shields were made of **a material of equivalent thickness.**

i. Fig. 21 presents a beryllium copper tube shield of the wrap-around type and shows an example of the variation in temperature around the circumference of such a shield. The temperature gradient of 72°C over the shields is typical and demonstrates the wide variation in the tube envelope temperature. It also serves to show that shield temperatures do not give an accurate indication of envelope temperature. In another test the plain copper wrap-around shield exhibited a tube base temperature of 128°C (See Fig. 22).

ii. A special wrap-around shield developed at C.A.L. (See Fig. 19C) was fabricated with two clips on the bottom to clamp the base of the tube for seal cooling and slotted circumferentially to permit more intimate contact with the tube envelope. This shield lowered the tube bulb temperature rise 15°C below that obtained with a simple



SECTION A—A

Fig. 21

TYPICAL TEMPERATURE GRADIENTS ON A
WRAP-AROUND TUBE SHIELD

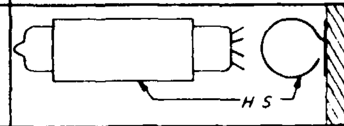
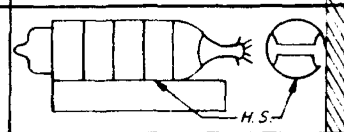
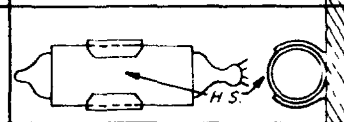
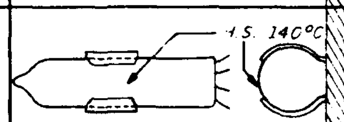
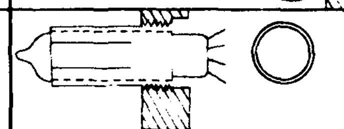
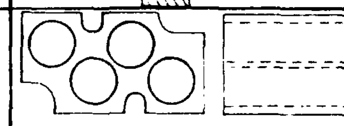
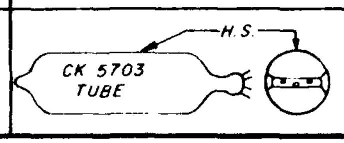
	TUBE SHIELD DESCRIPTION	°C ENVIROMENTAL TEMP	°C TUBE BASE	°C TUBE ENVELOPE (NEAR PLATE)	°C TEMP. RANGE OVER TUBE ENVELOPE	REMARKS (H.S.=HOT SPOT) LOCATION	RELATIVE MERIT IN ORDER OF EFFECTIVE COOLING
1	PLAIN WRAP-AROUND (COPPER)	23	128	103	——		E
2	SLOTTED WRAP-AROUND WITH BASE CLIP (COPPER) C A L	24	71	88	96 —— 61		C
3	FUSE CLIP WITH WRAP-AROUND AL	26	88	82	82 —— 60		C
4	FUSE CLIP (NO COVERING)	23	86	139	140 —— 56		D
5	SLOTTED CYLINDRICAL SHIELD (SCREW BASE) (AL.)	20.5	88	49	44 —— 41		A
6	METAL BLOCK (AL.)	24	82	RESULTS EFFECTED BY CONVECTION COOLING THIS TECHNIQUE PRODUCED THE LOWEST TEMPERATURE SEE TEXT			B
7	NO SHIELD TUBE VERTICAL IN AIR	24	103	163	163 ——		F

Fig. 22 SUMMARY OF SUBMINIATURE TUBE SHIELD EVALUATION

wrap-around shield (see Fig. 22).

iii. Tests performed with a fuse clip type mounting 7/16 inch long made of beryllium copper (see Figs. 18 and 22), showed a seal temperature of 86°C while the top of the tube envelope between the two fuse clips was 140°C. The temperature half way down the clip at the point of contact with the envelope was 56°C.

The high thermal gradient of 54°C between the tube envelope hot spot (above the fuse clips) and point of contact with the fuse clips indicates that the fuse clip type shield does not evenly cool the tube and that severe glass envelope gradients are produced. However, the tube base temperature with the fuse clip mounting was 86°C and good overall cooling was provided by conduction to the sink, plus free convection and radiation from the tube envelope.

iv. Tests performed with a combination aluminum wrap-around tube shield (see Fig. 22) mounted in a beryllium copper fuse clip indicated a 5.5°C rise in tube base temperature over that obtained with a simple clip. This is believed to be caused by the interference of the aluminum shield with free convection and radiation. The temperature of the shield over the hot spot was 86°C, while half way down the fuse clip the temperature was essentially the same. This indicated that the aluminum wrap-around shield reduced the individual hot spots by distributing the heat more evenly over the surface of the tube envelope but did not increase the conduction heat flow through the fuse clip. While the seal temperature was increased, the use of the wrap-around shield overcame the primary disadvantage of the fuse clip type in that the severe thermal gradients on the envelope were eliminated.

v. The aluminum cylinder type shield with the screw base and vertical slots developed at C.A.L. (see Figs. 19f and 22) was evaluated with the shield operating in the vertical position. The temperature of the tube envelope ranged from 44 to 41°C, while the temperature at the bottom of the shield in the direction of heat flow was 39°C and the tube base temperature was 88°C. With the tube and shield in the horizontal plane, the top of the shield away from the base plate was 44°C, and the tube base temperature was 87°C. Since the shield temperature was not significantly affected by mounting position, it was concluded that the predominate cooling modes were radiation and metallic conduction.

vi. The block type of shield was first tested with the tubes suspended in the holes and not contacting the metal. It was found that the tube base temperature was 82°C, the temperature on the outside of the block opposite the center (hottest point on the tube) was 32°C, and a little farther away it was 30°C. The envelope temperature was not measured because it was believed that this test was not conclusive, since the "chimney" effect provided convective cooling. With the tube molded in "Silastic", the temperature opposite the plate on the outside of the tube block was 33°C, and the tube base was 41°C. This is not considered to be conclusive evidence that the tube block is superior to other shields and work is continuing.

vii. When mounted vertically in free air, the tubes used in the above tests exhibited a base temperature of 103°C and an envelope temperature of 163°C. This is 40 to 50°C hotter than the equivalent miniature tube at the same power input.

(g) Conclusions

The findings of this evaluation are summarized in Fig. 22.

i. The fuse-clip type shield in conjunction with a flexible wrap-around shield is superior to the wrap-around shields for a given metal thickness, since two short parallel cooling paths are provided by the clip. Further, tube replacement is easier with the clip type shields.

ii. The cylindrical shield seems to be superior to all other shields.

iii. Under equivalent conditions, the envelope temperature of a subminiature tube with a good thermal shield will actually be lower than that of its miniature tube counterpart with a conventional shield.

It is realized that these tests omitted several recently developed shields and that these findings are not entirely conclusive, but only representative of the several tube shields evaluated.

(h) Design Notes on Subminiature Tube Shields

i. Base cooling of flat press-type subminiature tubes is not provided by most subminiature tube shields. It can be accomplished by incorporating springlike clips to contact the press base (see Figs. 19b, c, d, & e). With such a configuration some additional cooling can be achieved by conduction from the base of the tube. The degree of cooling obtained depends upon the heat conducted by the tube leads inside of the tube and the contact resistance between the clip and the base. Base cooling is recommended in instances wherein the tubes are operated near their maximum ratings or when the optimum cooling is required.

ii. Several tests to determine the effectiveness of blackening the inside of tight fitting tube shields were conducted. It was found that no significant temperature reductions were obtained. This was believed to be due to the increased thermal resistance of the blackening materials. Thin black oxide films were later used and no improvement resulted. Apparently the resistance of the blackening materials offsets the gain obtained by increasing the emissivity.

iii. Shields should be made of a highly conductive material, preferably copper or aluminum. The shield metal should be thick to provide a low resistance path. Also, the shield should be designed so as not to "crowd" the heat flow.

iv. Spring materials are usually necessary for tight fitting tube shields. Unfortunately, such metals are relatively poor conductors. For example, beryllium copper is only 20% as effective as pure copper. The use of spring metals can result in temperature gradients in the shield unless large thicknesses are used. A compromise can usually be achieved by utilizing pure soft metals in contact with the tube envelope and compressing the shield with an external spring.

Fig 23- Shield Configurations

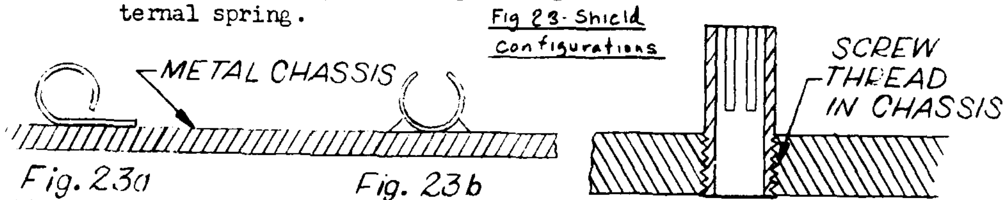


Fig. 23a WRAP-AROUND TYPE SHIELD

Fig. 23b FUSE CLIP TYPE SHIELD

Fig. 23c CROSS-SECTION THROUGH CYLINDRICAL SHIELD

v. A brief analysis of three typical tube shield types follows:

Fig. 23a shows an end view of a conventional shield. Since heat enters the shield radially over the entire surface, it can be seen that the shield carries ever increasing heat, starting from the open side or edge, and proceeding counterclockwise around the circumference. A better shield might be similar to that shown in Fig. 23b. Here the heat is transferred by two parallel paths which offer less resistance. Fig. 23c shows a shield which can provide very effective cooling. Note again that if the conductive path to the cooled surface is to be effective, there must be a good metallic bond between the shield and the surface. Further, the tube can be maintained at a permissible surface temperature only if the shield can transfer the heat gained from the tube to a sink at lower temperature.

c. Unshielded Tubes

Vacuum tubes larger than the subminiature types can be operated satisfactorily without shields if adequate convection cooling is provided and if the tube can "see" cooler surfaces (radiation). Convection cooling can be increased by providing "chimney" effect devices to guide the air flow. Sub-mounting in conjunction with a "sink strainer" will also aid convection cooling (See Fig. 24).

3. Resistors

Most resistors for electronic circuitry have been designed for natural cooling in free air. Resistors differ from vacuum tubes in that almost any power can be dissipated in a given resistor provided adequate cooling is present. Thus, with increased cooling, resistors can be operated successfully at increased ratings. The dissipation rating of a given resistor will therefore vary, dependent upon its environment. Resistor deratings are based on the maximum operating temperature and are published by resistor manufacturers. This section does not include such information.

The average 1/2 watt resistor in conventional equipment will reject approximately 40 per cent of its heat by free convection, 10 per cent by radiation, and 50 per cent by conduction through the leads. There is not much a designer can do to increase the cooling by radiation and natural convection other than to reduce ambient temperature. The conduction cooling of resistors can, however, be greatly increased. With small resistors, as noted above, considerable conduction cooling can take place through the leads. Further, with a 1/2 watt resistor it has been found that a 36°C rise above ambient temperature was obtained with zero length leads connected to a sink (ref. 13). With leads one inch long, the rise was 51°C. Correspondingly, the lead length had a greater effect when larger resistors were tested. Therefore, it is suggested that larger diameter leads be used with resistors and that the lead length be minimized. If possible, the leads should be thermally grounded to the chassis.

The body of the resistors should also be in contact with a metallic chassis or sink. Clamping to the chassis has been found to be very effective. The width of the clamp is not as important as the fact that, by clamping, the resistor body is in

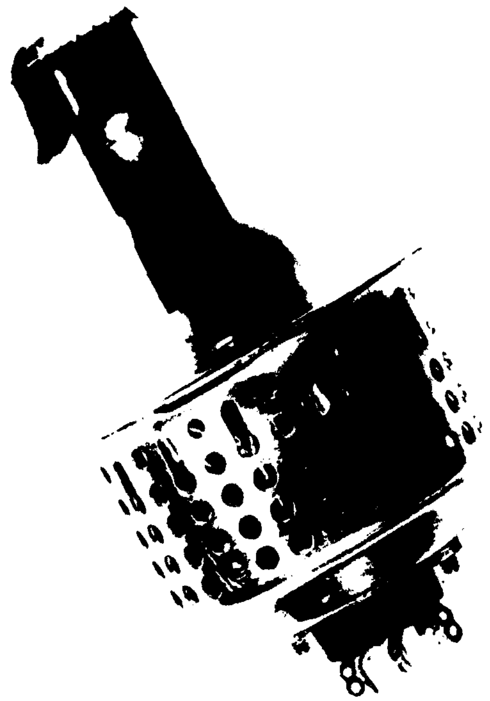


Fig. 24

SUBCHASSIS MOUNT USING A SINK STRAINER

intimate contact with the chassis. Further, it is desirable that the thermal conductivity of the resistor insulation be increased. This will aid all types of cooling.

4. Iron Core Inductors

Like resistors, iron core transformers and chokes can be cooled effectively by conduction cooling. The thermal resistance in the core is usually low and the hot spots in the core can be reduced by cooling the external surface of the core. The core should be thermally bonded over a wide area to a heat conducting support or chassis connected to a sink. Care must be exercised to prevent eddy current losses due to possible lamination short circuits at the bond.

Increased cooling, together with size and weight reduction, can be obtained by inserting metal heat conductors into the laminations and windings. In one instance, the temperature rise was reduced by 20°C and the weight was reduced from 18 to 8 pounds by the utilization of this technique (See Fig. 25).

D. NATURAL METHODS OF COOLING ASSEMBLIES

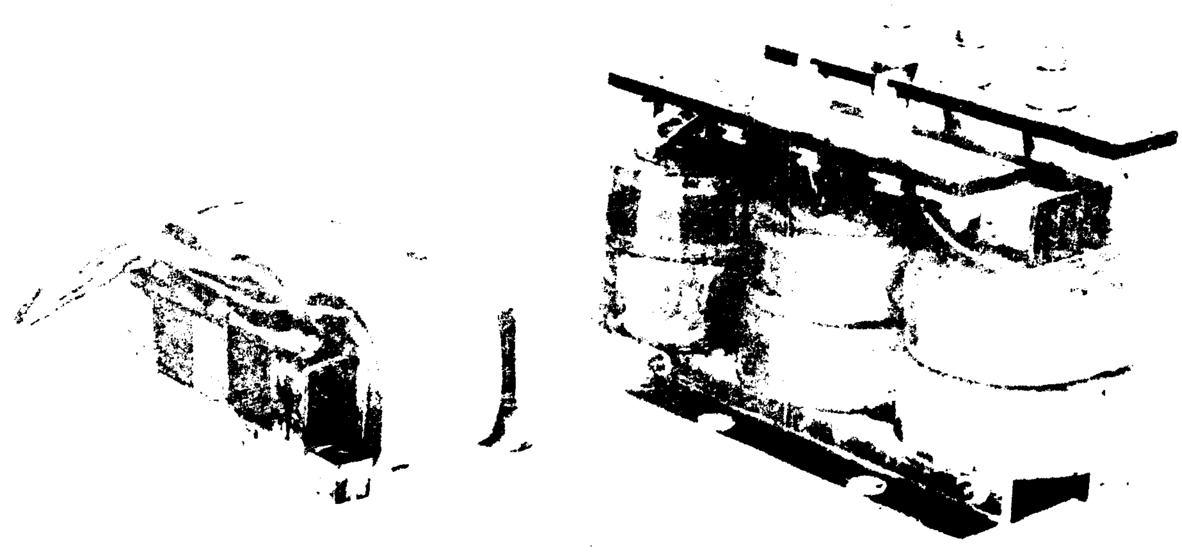
The thermal design of individual assemblies should be based on two fundamental concepts. The cooling technique should be such that a given subassembly rejects a minimum of heat to its neighbors and, a heat removal termination which is thermally matched to a "sink connection" must be incorporated. For example, several subassemblies developed at this Laboratory were provided with heat "sink connectors" as well as electrical connectors and the temperature ratings of these devices were based on the temperature of the heat removal studs at the thermal connector (See Fig. 26)(Ref. 14). A typical subassembly had a heat concentration of 2.1 watts/in.^{3}, a unit heat dissipation of 1/2 watt/in.^{2} and was rated for 150°C (500 hrs.life) or 125°C (5000 hrs. life) maximum "sink connector" temperature.

The primary cooling mode must be selected so as to provide a path of low thermal resistance from the heat producing parts to the sink. Radiation cooling is seldom used as a primary mode, since excessive temperatures usually result and the heat is rejected into nearby subassemblies. Convection cooling cannot be very effective unless large cooling areas and wide part spacings are available. This is seldom the case with miniaturized equipments. Conduction, therefore, appears to be the most desirable natural heat removal means. Further, the heat can readily be directed along a desired path.

Conduction cooling can be effective in two basic types of assembly construction, viz., those units using plastic embedment and metal chassis.

1. Metallic Conduction Cooling

Metallic conduction is one of the most satisfactory modes of heat



Courtesy RAYTHEON MFG CO

Fig. 25

MINIATURIZED TRANSFORMER USING METALLIC CONDUCTION COOLING — *CONVENTIONAL TRANSFORMER*

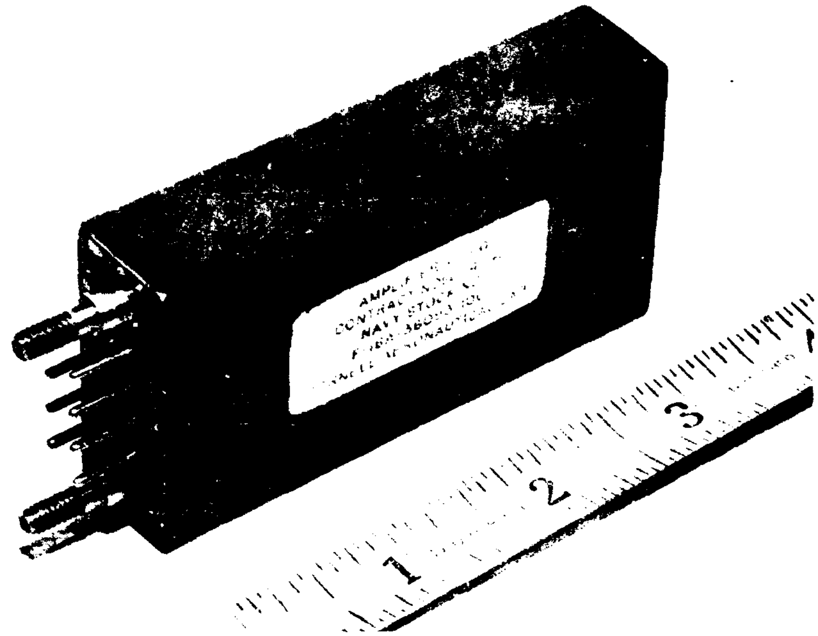


Fig. 26
VIDEO AMPLIFIER SUBASSEMBLY

transfer from heat producing parts to the "sink connector" in a miniaturized subassembly. As previously stated, metals have high thermal conductivities and it is therefore advisable to use metal tube shields and other metal parts in the thermal circuits. Several typical configurations follow:

a. Metal Block Chassis

This type of construction (see Fig. 27) involves a die cast or a machined metal block chassis with the tubes inserted in cylindrical holes near the cooled surface. The metal block forms the outside of the case of the subassembly. Case wall thicknesses of 0.125 inches are commonly used and thicknesses of 0.25 inch have been used in equipments of high heat concentration. In this instance, the non-heat producing parts are mounted in the center of the block. They may be placed at an edge of the block if desired. For maximum temperature operation it is advisable to thermally isolate the temperature sensitive parts from the heat producing parts. However, it is also necessary that the temperature sensitive parts be thermally grounded by a separate path so that they will operate at the lower temperature level of the sink connector. Note the difference between the good and poor designs.

b. Metal Chassis

Metal cases and chassis are commonly used. A typical design is presented on Fig. 28a (ref. 14). These subassemblies utilized 0.030 inch thick copper cases and 0.020 inch thick beryllium copper subchassis which were spring loaded against the inside of the cases to accomodate expansion. The tube shields were bolted to the subchassis near points of contact with the case so that the thermal path was as short as possible. With a heat concentration of 2.1 watts per cu. inch, the temperature gradients between the base of the tube shields and the heat conducting studs ranged from 2 to 17° C.

Many variations of this type of construction are possible. The thermal design can be approximated by utilizing the data presented under Item B. Unfortunately, the temperature gradients across joints, rivets, etc., cannot be accurately predicted. It is recommended that each configuration be simulated and the temperature gradients across the discontinuities be measured (see notes on contact resistance, Sec.V).

Hermetic sealing is frequently used to alleviate undesirable environmental effects. In such instances it is advisable to inert the subassembly with a gas having a high thermal conductivity such as hydrogen or helium. An additional reduction in the thermal gradient can thus be obtained.

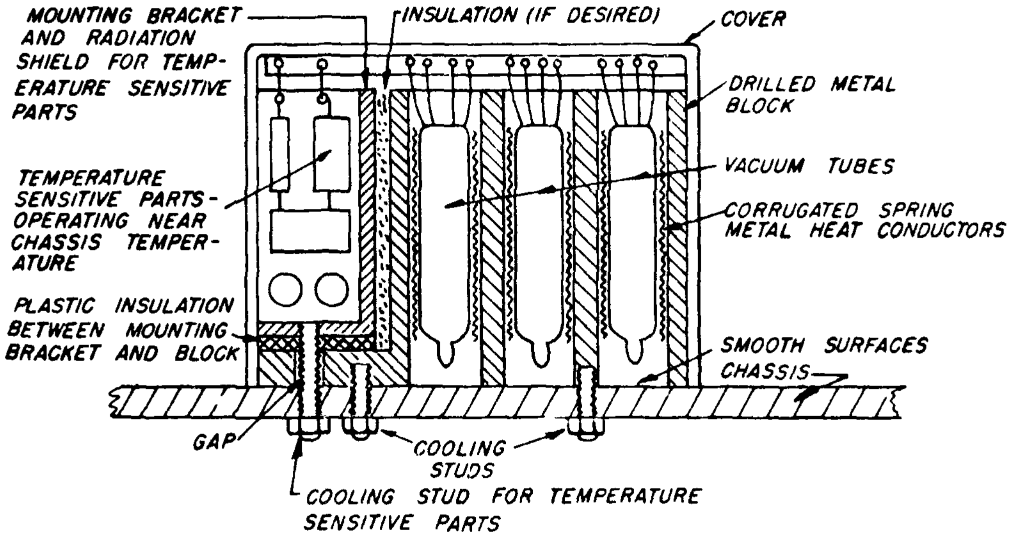


Fig 27a RECOMMENDED CONSTRUCTION PRACTICES

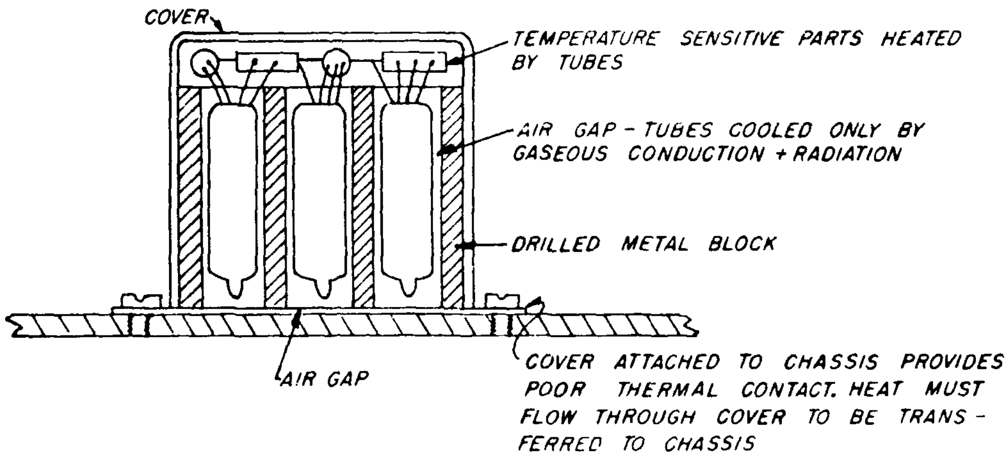


Fig 27 b POOR DESIGN

Fig. 27 DRILLED METAL BLOCK CONSTRUCTION

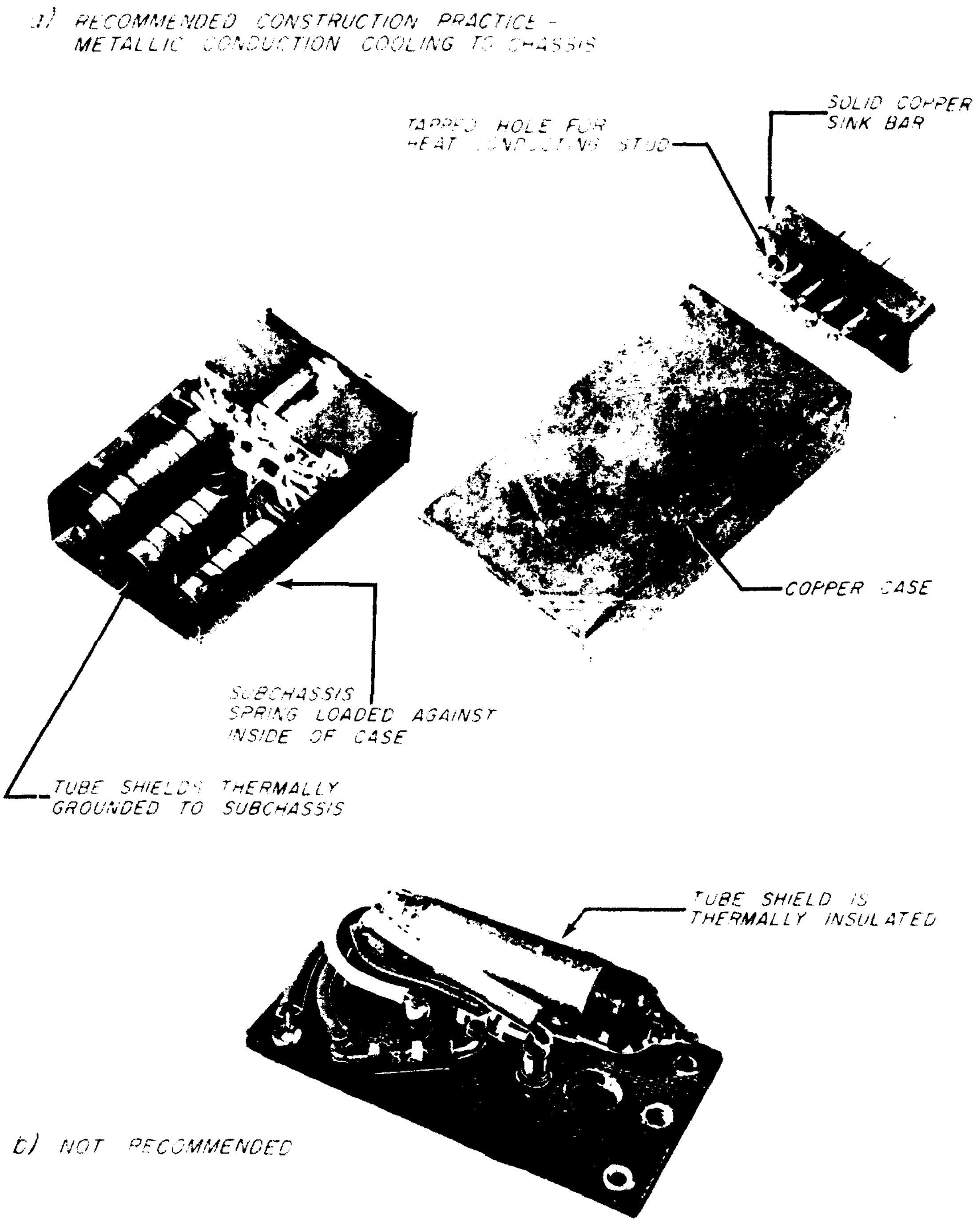


Fig. 28
CONSTRUCTION METHODS

c. Nonmetallic Chassis

Chassis of plastics and similar nonmetallic materials are not recommended. This type of construction can only be used for devices which dissipate extremely low power such as hearing aids. Fig. 28b shows an example of this type of construction. The tube will operate much cooler without the shield. Even so, when operated without the shield, in several instances the radiation from the tube charred and blackened the phenolic chassis.

2. Plastic Embedment

Heat transfer in embedded subassemblies is primarily by conduction through the plastic in conjunction with some metallic conduction in the wiring. A number of such subassemblies incorporate built-in metal heat conductors and are actually cooled by metallic conduction cooling. The thermal design of this type of subassembly should be based on metallic conduction alone, with the plastic serving only as a structural medium. This construction method should be used above 0.25 watts/cu.in. The following is limited to those types which rely on heat conduction through the plastic as the primary cooling mode.

In general, due to the poor thermal conductivity of plastics, extreme care must be utilized in designing a potted subassembly. Embedment materials which can withstand peak temperatures in excess of 165°C are not currently available. With heat concentration of the order of 0.5 watts/cu.in., excessive temperature gradients can easily occur, leading to mechanical fractures in the plastic and failure of electronic parts. For subassemblies having heat concentrations less than 0.25 watts/cu.in. plastic embedment will provide excellent heat removal. Further, the power handling capacity of conventional composition carbon resistors can be increased by plastic embedment. Excessive hot spot surface temperatures of resistors can thus be avoided. There is a need for high temperature embedment material having the highest possible thermal conductivity, together with low electrical conductivity. Until such a material is available, the applications of plastic embedment should be limited to electronic equipment having low heat concentrations.

In a typical instance, four type T-2 diodes were embedded with casting resin to form a cylinder 5/8" diam. x 1 3/4" long (ref. 15). Difficulty was experienced with cracked and overheated resin at 3 watts total heat dissipation. It was necessary to incorporate aluminum particles in the resin in order to achieve satisfactory thermal performance. The addition of fine metal particles will increase the thermal conductivity of embedment materials, depending upon the metal, the quantity involved, and the particle size.

E. THE PLACEMENT OF PARTS WITHIN ASSEMBLIES

Major thermal benefits can be achieved through the judicious placement of electronic parts within assemblies. In general, it is desirable to locate the heat sources as near as possible to the coolest surface. This will provide the shortest thermal path from the source to the sink, together with the minimum thermal gradient. In order to obtain maximum heat transfer, heat producing parts cooled by radiation and conduction should always be mounted with their major axes parallel to cooled surfaces. When convection cooling is utilized, heat producing parts should be mounted with their longer dimensions vertical. For example, vacuum tubes should be vertical because, in general, they will operate cooler than when mounted horizontally.

When heat producing electronic parts, cooled by convection and radiation, are mounted in closely spaced groups, much mutual heat transfer can occur by radiation, gaseous conduction and decreased individual convection cooling. Under these conditions, excessive operating temperatures can easily be obtained. Such arrangements can be very undesirable.

When it is necessary to group such heat producing parts together, metallic conduction cooling is recommended. Conduction cooling will greatly reduce the thermal interaction and will permit any practical spacing desired. If metallic conduction cannot be provided, then the parts should be arranged horizontally to form a "bank" of minimum height. If vertical stacking is necessary, then the parts must be staggered.

Resistors must be derated when mounted in groups. Fig. 29 illustrates the per cent of single unit rating vs. the number of resistors in the group. Separate curves are given for five different spacings and, in any group of three or more, the spacing between resistors is identical. Two percentage scales are shown; one for free air and the other for a mesh enclosure.

Several recommended techniques are presented in Fig. 30. Note that a polished radiation shield which is thermally bonded is provided between the rectifier tube, resistor and temperature sensitive parts. A cylindrical tube shield which is thermally grounded is placed around another vacuum tube to obtain guided convection and part protection. The selenium rectifiers are above a shield and mounted with their fins in the vertical position.

F. NATURAL METHODS OF COOLING ELECTRONIC EQUIPMENT CASES

Electronic equipment cases are occasionally designed to dissipate their heat to their environment by convection and radiation. Such a cooling technique is practical provided the unit heat dissipation is low, or the ambient air temperature and the temperature of surrounding walls is low.

Figs. 31a & 31b present the predicted temperatures for given case configurations. Section IV presents data for computing the thermal capacity of this type of equipment case.

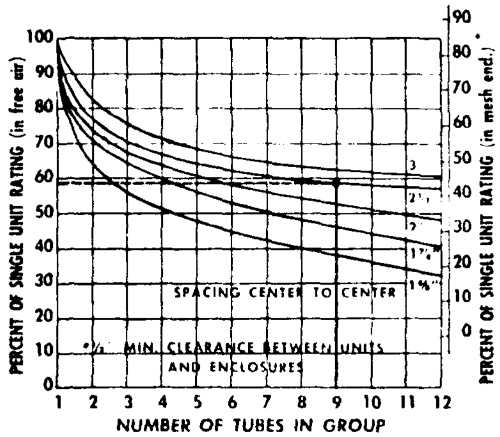


FIG. 29A
GROUP MOUNTING OF RESISTORS

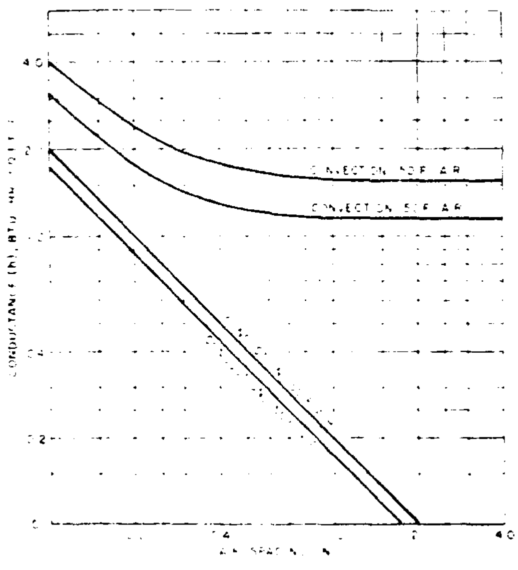


FIG 29B
AIR SPACE CONDUCTANCE CURVES

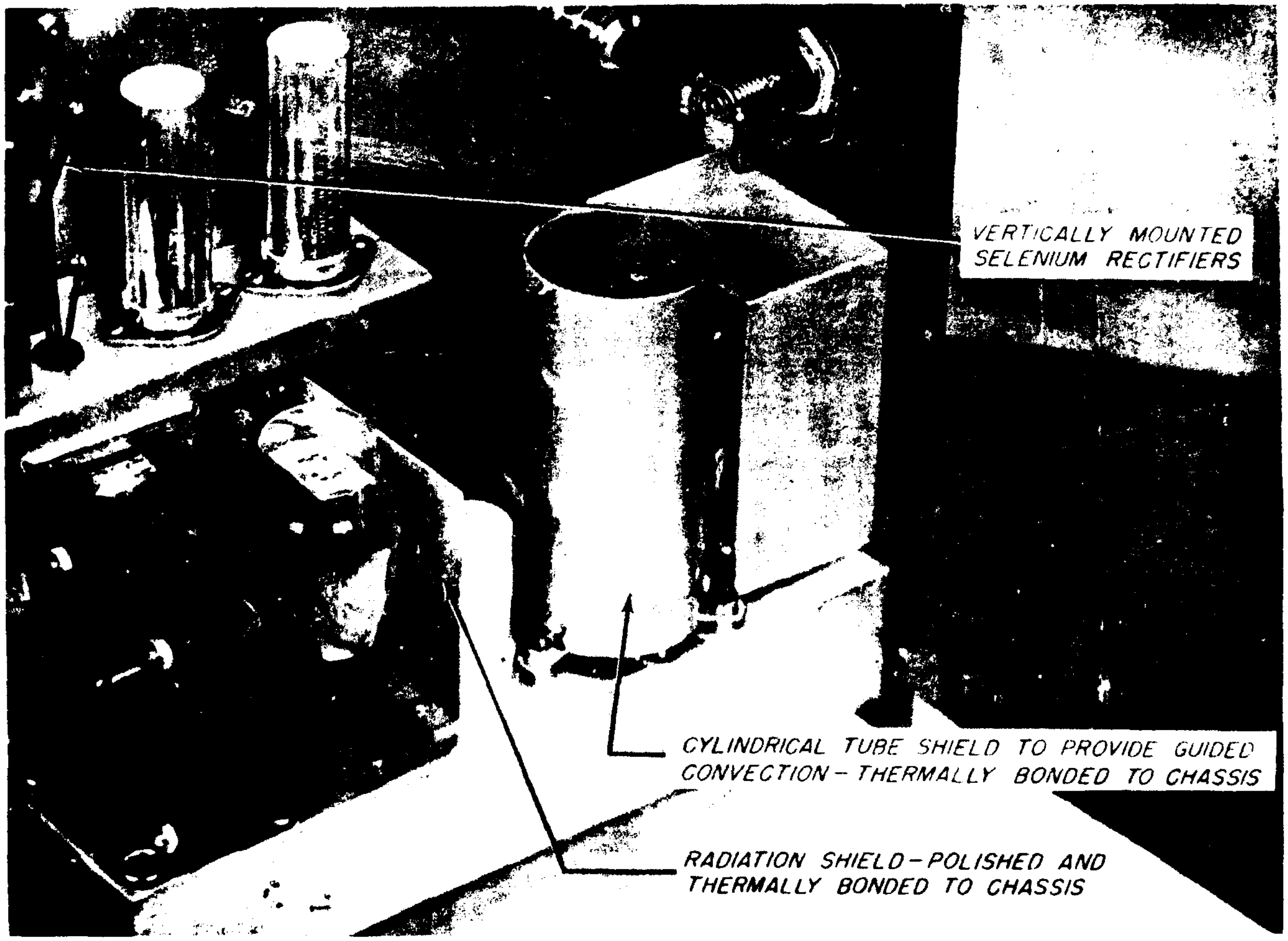


Fig. 30

RECOMMENDED CONSTRUCTION PRACTICES IN CONVENTIONAL EQUIPMENT

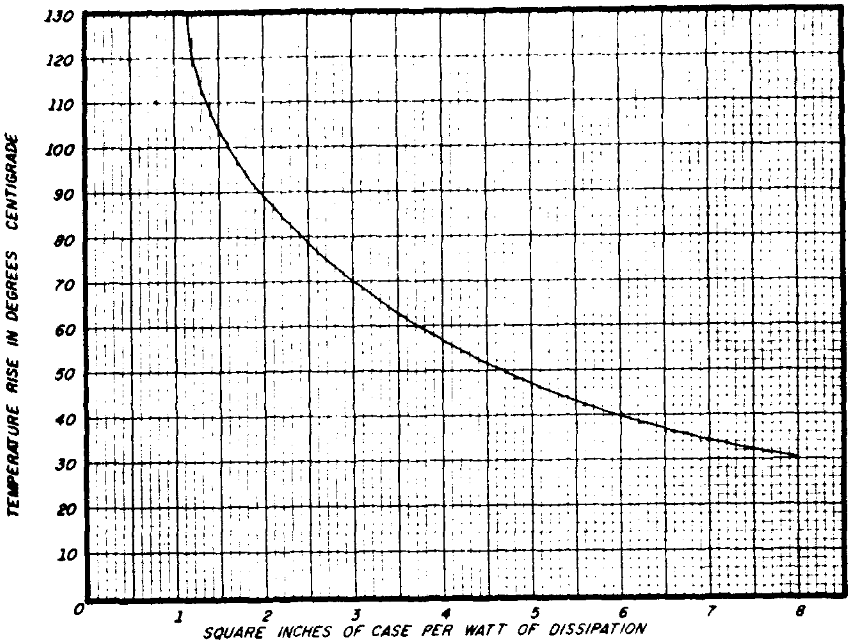


Fig. 31a—TEMPERATURE RISE OF TYPICAL ETCHED ALUMINUM CASE IN STILL AIR AT ROOM TEMPERATURE

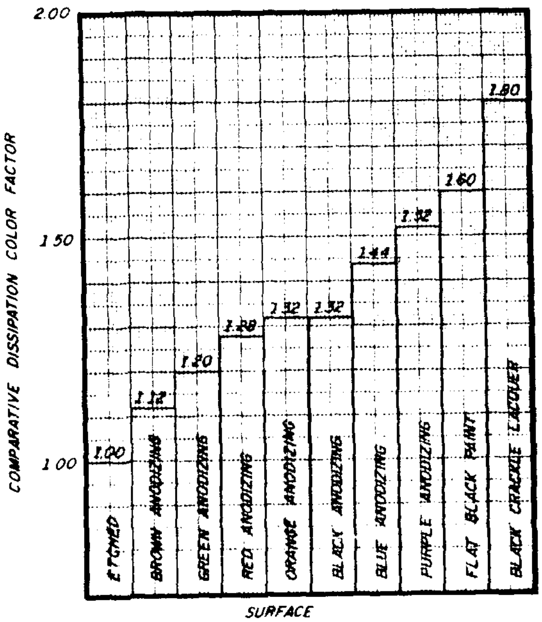


Fig. 31b—COMPARISON OF DISSIPATIONS INTO STILL AIR OF ALUMINUM SURFACES

VI. FORCED AIR COOLING

A. GENERAL THEORY

Forced air cooling of a heated surface is a more effective method of heat removal than free convection. Increasing the air velocity past a heated surface results in a decreased resistance to heat transfer across the air film and increased cooling which may be several times that for free convection.

Like free convection, forced convection is a function of several variables and the general equation includes the three dimensionless groups: the Nusselt, Reynolds and Prandtl numbers. The equation takes the form:

$$Nu = C(Re)^m (Pr)^n \tag{32}$$

The free convection equation included the Grashof number. For forced convection, however, the Grashof number is replaced by the Reynolds number which can be defined by:

$$Re = \frac{LV\rho}{\mu} \text{ or } \frac{LG}{\mu} \tag{33}$$

where:

- L is a characteristic dimension
- V is the air velocity
- ρ is the air density
- μ is the air viscosity
- G is the mass velocity (pounds air flowing across a unit area per unit time)

As in free convection, it is sometimes more convenient to use the Btu.-pound-foot-hour-degree F system of units in calculations. These are listed in Appendix A.

In general, the type of flow, either streamline or turbulent, is indicated by the magnitude of Reynolds number. For example, for a fluid flowing within a pipe, if the Reynolds number is above 3100, the flow is turbulent and, if below 2100, the flow is streamline. The transition from streamline to turbulent flow occurs in the region from 2100 to 3100. The mechanism of heat transfer in streamline flow differs from that in turbulent flow, the latter producing the higher rate of heat transfer. Since forced air cooling of electronic equipment usually involves turbulent flow, this discussion does not include streamline flow heat transfer.

1. Flow Within Tubes or Pipes

The equation for the film coefficient for turbulent flow of any fluid within a pipe, except those having viscosities more than twice that of water, is given by equation (34), (ref. 3). For flow within tubes, turbulent flow predominates when the Reynolds number is greater than 3100.

$$h_c = 0.023 \frac{k}{D} \left(\frac{DV\rho}{\mu}\right)^{0.8} \left(\frac{c\mu}{k}\right)^{0.4} \tag{34}$$

Here the characteristic dimension is the inside tube diameter D. For the Prandtl number varies little with temperature, and equation (34) may be simplified by substituting 0.69 for the Prandtl number which results in:

$$h_c = 0.0198 \frac{k}{D} \left(\frac{DV\rho}{\mu}\right)^{0.8} \tag{35}$$

Thus, for air being either heated or cooled within a pipe (exchanging heat with the walls), the coefficient of heat transfer may be approximated by equation (35). For other gases, equation (34) should be used.

For ducts of non-circular cross-section, the equivalent diameter D_e for use in these equations is defined by:

$$D_e = \frac{4 \times \text{cross-sectional area}}{\text{cross-sectional perimeter}} \tag{36}$$

2. Flow of Gases Parallel to Plane Surfaces

The film coefficient for air flowing parallel to smooth plane surfaces is given by:

$$h_c = 0.055 \left(\frac{k}{L}\right) \left(\frac{LV\rho}{\mu}\right)^{0.75} \tag{37}$$

Here L is the length of the surface and is limited to two feet, even if the length of the surface is greater (ref. 11).

3. Flow of Air Across Wires and Cylinders

The equation for the film coefficient across a single wire or cylinder takes the form:

$$\frac{hD}{k} = b\left(\frac{DV\rho}{\mu}\right)^{n} \tag{38}$$

TABLE 6
(From References 16 and 17)

Constants for Use in Equation (38) for Round Cylinders

$\frac{DV\rho}{\mu}$	b	m
0.4 - 4	0.891	0.330
4 - 40	0.821	0.385
40 - 4,000	0.615	0.466
4,000 - 40,000	0.174	0.618
40,000 - 400,000	0.0239	0.805

The constants b and m, are dependent on the magnitude of the Reynolds number and are given in Table 6. The air properties should be evaluated at the mean of the arriving air and surface temperatures, and h is defined for the difference in these temperatures. The arriving air velocity is the reference velocity.

4. Flow of Air Across Cylinders of Square Cross-Section

The equation for the film coefficient for air flowing normal to the axis of cylinders of square cross-section is similar in form to that of cylinders of circular cross-section. The constants b and m depend on the orientation of the cylinder with respect to the direction of airflow and are given in Table 7 below:

TABLE 7
(From Reference 17)

Constants for Use in Equation (38) for Square Cylinders

Cross Section	Reynolds No.	b	m
→ □	5,000 - 10,000	0.0921	0.675
→ ◇	5,000 - 100,000	0.222	0.585

Note that the constant b for the flow parallel to a diagonal is more than twice that for flow parallel to a side. The characteristic dimension L is the diameter of a circular tube of equal cross-sectional perimeter and b and m are defined as for flow across wires and cylinders.

5. Flow of Air Across Banks of Tubes or Circular Cylinders

The correlation of the data for the flow of air across banks of circular cylinder tubes takes the same form as equation (38). The Reynolds number, $DV\rho/\mu$, may be written as DG/μ where G is the mass flow rate in pounds per hour per sq. ft. of cross-section normal to the air flow. The value of G in the correlation of the data is that at the narrowest cross-section between the vacuum tubes whether or not the minimum area occurs in the transverse or diagonal openings between tubes. Thus, Equation (38) becomes:

$$\frac{hD}{k} = b\left(\frac{DG}{\mu}\right)^m \qquad (39)$$

The constants b and m depend on the distance between the tubes, the tube diameter and the arrangement, i.e., staggered or in line. Table 8 gives the values of b and m obtained by Grimmison (See Ref. 17) to be used in Equation (38) for banks of tubes 10 rows deep. The term S_L and S_T are the longitudinal and transverse tube pitches respectively, as defined in Fig. 32.

Fig. 32

Tube Bundles with Tubes in Line and Staggered

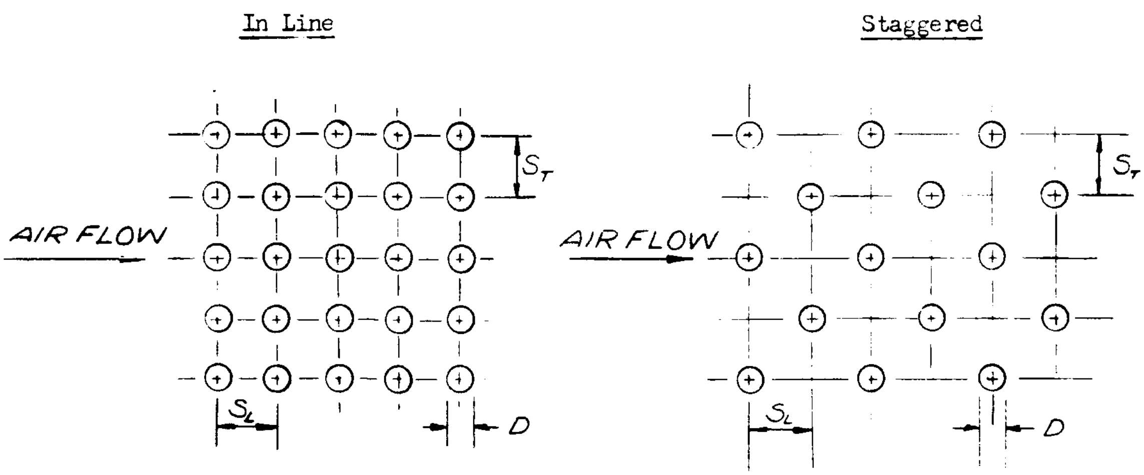


TABLE 8

Reynolds' Number Functions for Staggered Tubes

Values of b and n to be used in Equation (38) for Airflow Normal to Tube Bundles (Ref. 17) (10 row deep banks).

S_L/D	S_T/D							
	1.25		1.50		2.0		3.0	
	b	m	b	m	b	m	b	m
Tubes in Line								
1.25	0.348	0.592	0.275	0.608	0.100	0.704	0.063	0.752
1.5	0.367	0.586	0.250	0.620	0.101	0.702	0.068	0.744
2.0	0.418	0.570	0.299	0.620	0.229	0.632	0.198	0.648
3.0	0.290	0.601	0.357	0.584	0.374	0.581	0.286	0.608
Staggered Tubes								
0.6							0.213	0.636
0.9					0.446	0.571	0.401	0.581
1.0			0.497	0.558				
1.125					0.478	0.565	0.518	0.560
1.25	0.518	0.556	0.505	0.554	0.519	0.556	0.522	0.562
1.5	0.451	0.568	0.460	0.562	0.452	0.568	0.488	0.568
2.0	0.404	0.572	0.416	0.568	0.482	0.556	0.449	0.570
3.0	0.310	0.592	0.356	0.580	0.440	0.562	0.421	0.574

In general, increasing the number of tube rows in the direction of airflow produces increased turbulence towards the rear of the tube bundle. Hence, the heat transfer coefficients are greater for the tubes in the rear. Table 9 shows the ratio of the mean coefficient for N rows to that for 10 rows deep (see Ref. 3).

TABLE 9

Ratio of Mean Film Coefficient for N Rows Deep to that for 10 Rows Deep.

N	1	2	3	4	5	6	7	8	9	10
Ratio for Staggered Rows		0.73	0.82	0.88	0.91	0.94	0.96	0.98	0.99	1.00
Ratio for In-line Rows	0.64	0.80	0.87	0.90	0.92	0.94	0.96	0.98	0.99	1.00

6. Flow of Air Over Spheres

For airflow over spheres, the film coefficient is given by:

$$\frac{hD}{k} = 0.33\left(\frac{DG}{\mu}\right)^{0.6} \qquad (40)$$

7. General Heat Transfer Equation

The general heat transfer equation for a fluid flowing past a heated surface is:

$$q = h_c A \Delta t_m \qquad (41)$$

where:

A is the area of the heated surface
Δt_m is the mean temperature difference between the surface and the air.

Since the air increases in temperature when flowing past a heated surface, the final air temperature must be evaluated by a thermal energy balance and the mean temperature difference then approximated as the average air temperature minus the average surface temperature.

The amount of thermal energy absorbed by air is given by:

$$q = wc\Delta t \qquad (42)$$

where:

w is the airflow rate in pounds/hr.
c is the specific heat in Btu./(lb.)($^{\circ}$F)
Δt is the air temperature rise in $^{\circ}$F
q is the heat rate in Btu./hr.

For air, w is given by:

$$w = 60 \times cfm \times \rho \qquad (43)$$

where:

cfm is the airflow in cu.ft./min.
ρ is the air density in lbs./cu.ft.

The flow rate in cfm is equal to the average air velocity times the net cross-sectional area normal to the directional flow, or

$$cfm = VA \qquad (44)$$

V is in feet/min. and A in sq. ft.

The density of air is given by the following:

$$\rho = 2.7 \frac{p}{t + 460} \qquad (45)$$

where:

p is the absolute pressure in lbs./sq. in. (or barometer in in. of mercury/2.036)
t is in ^{o}F.

It is important to note that equations (41) and (42) are the two basic equations which enter into heat transfer problems involving forced convection.

8. Heat Transfer vs. Cooling Power Requirements

The power required to force air over objects and through ducts and heat exchangers varies directly with friction or pressure drop. High rates of heat transfer are brought about by high air velocities but high air velocities result in high friction. Hence, the price to be paid for high heat transfer rates is the relatively large cooling power requirements.

Ref. (8) presents design charts for the forced air cooling of banks of electronic tubes based on the minimum cooling power requirements. This work is useful in designing for optimum conditions and reference to it is recommended. Further, pressure drop charts, together with typical examples, are presented.

B. FORCED AIR COOLING DESIGN

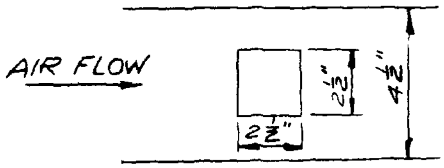
1. Use of Forced Convection Equations in Electronic Design

A relatively simple problem is presented in the following section to illustrate heat transfer calculations for air flowing over a small electronic box. The problem involves one in which the configuration is already given and not one of determining the best configuration. For more detailed information on optimum design methods, see Ref.(..).

a. Example 5

Use of Forced Convection Equations

An electronic box, 2 1/2 in. square by 4 in. high, is located within a 4 1/2 in. square passageway in an electronic assembly. The bottom of the box rests on the bottom of the passageway. A top view is shown.



Problem:
If 100 cfm of air at 49°C (120°F) is available and if the surfaces of the box are at 93.3°C (200°F), how much heat can be dissipated from the sides and top of the box by forced convection?

Area of passageway: $\frac{4.5 \times 4.5}{144} = 0.141$ sq. ft.

Arriving air velocity: $\frac{100 \text{ cfm}}{0.141} = 709$ ft./min.

or: $709 \times 60 = 42540$ ft./hr.

Approx. average film temp.: $\frac{200 + 120}{2} = 160^\circ F$

Properties of air at 160°F (from Table 18 appendix C)

$k = 0.0172$ Btu./(hr.)(ft.)(°F)
$\rho = 0.064$ lbs./cu.ft.
$\mu = 0.050$ lb./(ft.)(hr.)

Equivalent diameter of box = diameter of circle of equal perimeter

Perimeter of box = $2.5 \times 4 = 10$ in.

$$D = \frac{10}{\pi \times 12} = 0.265 \text{ ft.}$$

Reynolds number: $\frac{DV\rho}{\mu} = \frac{0.265 \times 42,600 \times 0.064}{0.050} = 14,400$

Film coefficient: Substituting in Equation (38) using constants from Table 7

$$\frac{h \times 0.265}{0.0172} = 0.0921\ (14,300)^{0.675}$$

$$h = 3.82\ \text{Btu./(hr.)(sq.ft.)}(^\circ F)$$

Area of box: Sides $4 \times 2.5 \times 4 = 40$ sq. in.

Top $2.5 \times 2.5 = 6.25$

Total 46.25 sq.in. = 0.321 sq.ft.

Temperature difference: Since h is defined for the difference between the surface and arriving air temperatures, the Δt is:

$$\Delta t = 200\text{-}120^\circ = 80^\circ F$$

Heat dissipation by forced convection:

$$q = 3.82 \times 0.321 \times 80 = 98\ \text{Btu./hr.}$$

or $$\frac{98}{3.413} = 28.8\ \text{watts}$$

Air temperature rise:

Air flow = 100 cfm x 0.068 x 60 = 408 lbs./hr.
Specific heat of air = 0.241 Btu./(lb.)($^\circ$F)
Air temperature rise = $\frac{98}{0.241 \times 408} = 1.0^\circ F$

b. Comparison with Free Convection

It is of interest to compare the dissipation by forced convection in Example 5 with that by free convection. The environmental temperature surrounding the box is assumed to be 120°F. The following calculations are based on the free convection chart (Fig.11).

Surface	Area Sq.in.	Significant Dimension	Film Coefficient Watts/ (sq.in.)(°C)	Watts Dissipation
Top	6.25	$\frac{2.5 \times 2.5}{2.5 + 2.5} = 1.25$	0.325	2.03
Sides	40	4.0	0.182	7.28
				9.31 Total

Thus, in this particular case, about three times as much heat is dissipated by forced convection.

c. Surface Temperature Variation

The equations for the film coefficient of air under forced convection all presume an isothermal surface. However, electronic parts, such as tubes, resistors and enclosing boxes, do not have isothermal surfaces, and considerable surface temperature variation may exist. This temperature variation is not only a function of the internal structure of a part such as a vacuum tube, for example, but may vary with the manner in which it is cooled. A small box containing electronic parts usually is hotter at the top than at the bottom. Since the unit dissipation is a function of the difference between surface and air temperature, an average temperature must be used. If the maximum or hot-spot temperature is used, an erroneous and too high a calculated heat dissipation will result.

There are no exact rules by which the average temperature of a heated part or container can be calculated. However, a reasonable estimate is usually satisfactory. Information on this matter is included in Section X. Also, Ref. (10) presents some data related to the temperature distribution over the surface of four vacuum tube types cooled by several methods.

It is possible to obtain an average temperature by direct experiment. Thermocouples can be placed on the surfaces of an object and an average temperature can be calculated on an area-weighted basis. Such a test should be conducted with whatever cooling method is to be used. With forced air convection, the cooling is most effective where the air is directed against the hottest surface.

2. Finned Surfaces

The effectiveness of forced air cooling usually can be increased by providing extended surfaces or fins over which the air is directed. The general theory is that the fins provide additional heat transfer surfaces which more than compensate for the small increased resistance to heat transfer offered by the metal of the fins.

The mathematics of fins is involved and will not be presented here. Excellent treatment of extended surfaces is given in Ref. (16).

There are several important general factors to consider. The extended surface should be of a good heat conducting metal. The fins should be either integral with the part or bonded to the part in perfect metal to metal contact so that there is a minimum of contact resistance. Short, rather thick, fins are more effective than long, thin ones. The temperature drop from the base to the tip of a long fin may be appreciable, and tends to make the fin less effective. Where applicable, weight and space requirements should also be considered.

3. Sealed Cases With Internal Forced Convection

Electronic assemblies in sealed cases with internal air circulation are frequently used with airborne electronic equipment. The cases usually employ an internal blower to provide forced air circulation over the electronic parts in order to transfer the heat to the case effectively. There are a number of variations of such sealed cases, including integral heat exchangers and external blowers (Ref. 20 and 21).

The application of sealed units permits pressurization so that effective forced convection can be maintained at high altitudes. The film coefficient for forced convection varies with the air density to a power close to unity. Hence, at high altitudes, convection can be effective only if the case is pressurized. Also, gases having higher film coefficients than air might be used. Helium, for example, would have a film coefficient of the order of five times that for air based on the same mass flow rate, lbs./(hr.)(sq.ft.). Thus, the use of gases other than air offers increased cooling potentialities.

The use of sealed cases has largely been confined to relatively large containers. However, the advent of extremely small blowers should make possible the application of forced convection to relatively small sealed cases.

C. PRESSURE DROP AND COOLING POWER REQUIREMENTS

1. General

An important consideration in forced air cooling is the pressure drop required to force the air through the ducts and passageways and over the parts or cases to be cooled. A fan or blower * must furnish this energy which, in turn, is usually supplied by an electric motor. The power requirements may become excessive and the electronicdesigner must realize that forced air cooling requires the expenditure of power. Further, this power is ultimately dissipated as heat which may add to the overall cooling problem. Detailed treatment of this rather complex subject is not included herein, and only important general considerations are reviewed.

* Note: The terms "fan" and "blower" are used interchangeably.

2. Cooling Power Requirements

As previously mentioned, high air velocity results in high film coefficients. To obtain air movement a fan or blower is required to increase the pressure of the air until it is equal to the pressure drop or resistance of the system. The power required for a given system is a function of the flow rate and system pressure drop and is given by:

$$\text{Air horsepower (AHP)} = 0.000157 \text{ cfm} \times \text{static pressure drop} \tag{46}$$

where:

cfm is the air handled in cu.ft. of air per minute
The static pressure drop is in inches of water.

Since the efficiency of a fan is less than unity, the required fan horsepower is given by:

$$\text{Fan horsepower (FHP)} = \frac{0.000157 \times \text{cfm} \times \text{static pressure drop}}{\text{fan static efficiency}}$$

where:

the fan static efficiency is expressed as a decimal, or

$$\text{Watts input to fan} = \frac{0.117 \times \text{cfm} \times \text{static pressure drop}}{\text{fan static efficiency}}$$

3. Fluid Friction & Pressure Drop

The fan must be selected to deliver the required air flow at a pressure equal to the sum of all the pressure losses of the system. In addition to the resistance to flow offered by the electronic parts, whether heated or not, the pressure loss or drop must be estimated for each item in the flow path, including all ducts, elbows, enlargements and contractions. Further, if a heat exchanger is used to cool the air, this resistance must also be considered. Resistance to flow within tubes or pipes and ducts, elbows and over pipe bundles can be calculated. For this, there are several available references, such as Ref.(3). It is difficult to calculate the pressure drop through densely packaged electronic equipments. This resistance probably is a major percentage of the total and it appears that it can be accurately determined only by test. Such test procedures and methods are described in Ref. (6).

The resistance to flow is proportional to the square of the air velocity. Thus, for a fixed duct and electronic package design, the resistance is proportional to the square of the air flow. For example, in a fixed design, if the air flow is doubled, the resistance quadruples. Also, since the power required is proportional to airflow times resistance, the power becomes eight times as great.

4. Conversion of Cooling Power Into Heat

The power required by the fan, that is, the power defined by equations (47) and (48), goes into the air to increase its pressure, velocity and temperature. It is conservative to consider that all of the input power is expended in increasing the air temperature. The approximate temperature rise of the air due to fan power input:

$$\Delta t = \frac{177 \text{ x fan horsepower}}{\text{cfm x density of air}} \tag{49}$$

where:

Δt is in ^{o}F
air density is in lbs./cu.ft.

For low pressure systems this temperature rise is usually small.

The losses of the motor, expressed in terms of power, are dissipated in the form of heat. If the motor is located outside of the air stream, such as one driving a centrifugal fan, then the losses of the motor are dissipated as heat to its environment. Since motors must also be maintained at safe operating temperatures, ventilated motor spaces may be required to prevent, for example, excessive temperature rise. On the other hand, if the motor is located in the air stream as, for example, in the case of an axial flow fan, then the entire input to the motor is dissipated to the cooling air in the form of heat.

D. FANS AND BLOWERS

1. General

The terms "fans" and "blowers" are usually used interchangeably and, in this manual, these terms will be used to cover all air moving apparatus. Reference (6) contains a comprehensive section on fan performance and selection and only a general discussion is presented here.

There are two types of fans: the centrifugal fan and the axial-flow fan. The type to be selected for a specific cooling problem is dependent on several factors, such as airflow and pressure requirements, efficiency, speed, space, the air ducting system, noise and fan characteristics.

2. Centrifugal Fans

A typical centrifugal type fan is shown in Fig. 33. The three important parts are the housing containing the air inlet and outlet, the rotor containing the fan blades, and the external driving motor. The air enters the housing normal to the side and is discharged in a direction perpendicular to the axis of rotation.

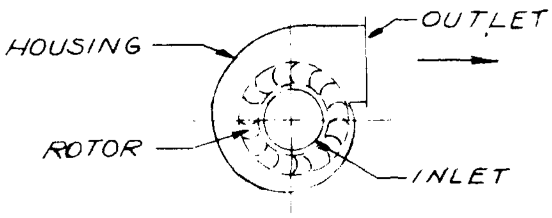


Fig. 33 Typical Centrifugal Fan

The centrifugal fan usually is more adapted to high pressure requirements.

3. <u>Axial-Flow Fans</u>

Axial-flow fans are of two types: the so-called propeller type, shown in Fig. 34, and a more efficient design shown in Fig. 35, commonly considered to be more truly an axial-flow fan.

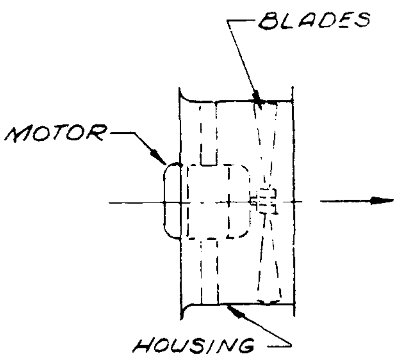


Fig. 34

Propeller Type Fan

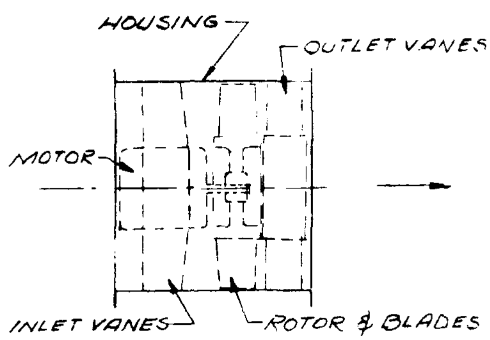


Fig. 35

Axial Flow Fan

There are several types of construction but all are typified by straight-through flow of the air. The axial-flow fan is of more efficient design and is a higher pressure fan than the propeller type.

4. Fan Characteristics

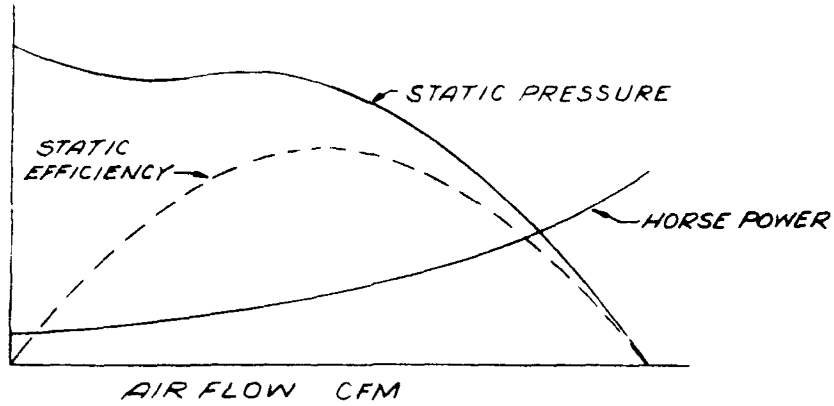


Fig. 36a Characteristic Curve of a Centrifugal Fan at Constant Speed

Fig.36a shows typical performance curves of a centrifugal fan running at constant speed. While there are various types of both centrifugal and axial-flow fans which have different characteristic curves, in a general way, Fig.36a is typical. With the outlet blocked there is no airflow and usually maximum static pressure. At maximum airflow or free delivery, the static pressure developed is zero. There is some point between these two extremes where efficiency is a maximum and, where power is important, the fan should operate at near maximum efficiency. This requires that optimum fan type, size and speed be selected to deliver the specified airflow and static pressure. In other words, the fan must be well matched to the system requirements.

5. Interrelation Between System Resistance and Fan Performance

In order to show the relation between the duct system resistance characteristics and the fan performance, Fig.36b, is presented:

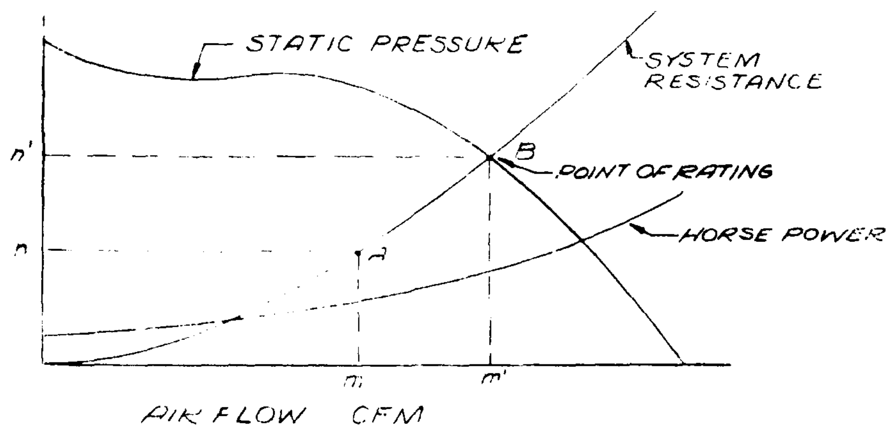


Fig. 36b Fan and Duct System Performance

First consider the "system" resistance curve. This is the static pressure - airflow characteristic of the forced air cooling system. For example, point "A" shows that at m cfm airflow through the system, the required static pressure is n inches of water at a given air density. The pressure drop varies as the square of the airflow rate so that the system resistance is a squared curve. The system rating point is that where the system resistance curve crosses the fan static pressure curve or, in this case, point "B". Thus, the airflow will be m' cfm at n' static pressure. It can be seen that it is important to select the fan so that its static pressure curve passes through or near the system resistance curve at the desired airflow and pressure near peak efficiency (if efficiency is important). Further, the speed and noise level must also be considered. Referring to the example in Fig. 36b, if point "A" were desired but a fan was used having characteristics as shown, the airflow actually delivered would correspond to that at "B" and the horsepower would be necessarily high. Hence, judicious selection of fans is important. The fan or blower manufacturer can be of great help in proper fan selection and it is suggested that his advice on difficult fan problems be solicited.

E. COLD PLATE HEAT EXCHANGERS

A cold plate heat exchanger is usually a cooled plate, chassis or panel on which are mounted the electronic assemblies and subassemblies. The plate may be cooled by forced air flowing through tubes which are in intimate contact with one side of the plate, or the plate itself may consist of a "sandwich" type heat exchanger through which the air is forced. The heat from the individual parts is usually transferred to the surfaces of the subassemblies by natural methods.

The inherent design requirements of a cold plate are such that electronic boxes and parts must be placed in intimate contact with the cold plate heat exchanger to provide a low resistance heat path from the metal boxes through the plate into the cooling fluid. This requires careful and, perhaps, expensive construction. The plates are particularly well adapted to applications wherein free air convection is poor due to low air density and radiation is reduced due to high environmental temperature. Thus, if hermetically sealed electronic components are mounted in good metal to metal contact with a cold plate, effective cooling can be obtained.

When air is used as the coolant in the cold plate, the air flow within the tubes or heat exchanger passages must be highly turbulent to decrease the resistance to heat transfer offered by the air film. This causes a relatively high air pressure drop with consequent high power requirements. Air plates described in Refs. 18 and 19 were designed for pressure drops as low as 0.1 psi and as high as 10 psi. It was concluded that 0.5 psi is the most practical, since duct sizes become excessive below and power requirements above the 0.5 psi drop. Design data including mathematical prediction methods are also presented. The actual performance of chassis of various sizes has been measured. Configurations and heat dissipations were within 15 percent of that predicted.

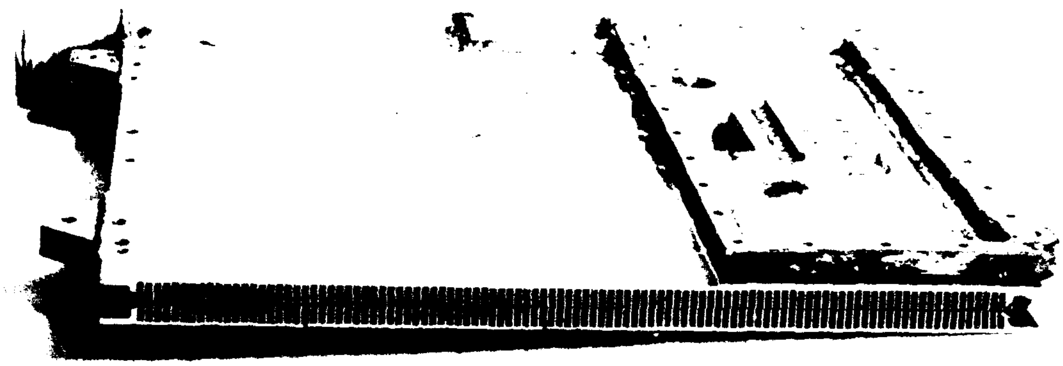
A low resistance path must be provided by conduction between subassemblies and the surface of cold plates. Thus, it is necessary that the cold plates and subassemblies have smooth and flat contact surfaces. If care is not exercised in this regard, the contact resistance to heat transfer can be excessive and the thermal advantages of the utilization of cold plates may not be realized. It has been found that it is necessary that cold plates be stiff and rigid so that no mechanical distortion will occur when the subassemblies are firmly mounted. The method of attachment of the subassemblies should provide maintenance accessibility and be such that component replacement may be accomplished with a minimum of effort.

The subassemblies may be attached to the plate by bars and through-bolts to provide good metal contact. Cold plates have been built with heat dissipation capacities ranging from 300 to 800 watts per sq. ft. of effective plate area. In one instance, the problem of temperature rise across the contacting surfaces between cooled chassis and component subassemblies was overcome through application of silicone grease. This method replaced any existing layers of air with a material having a thermal conductivity from 50 to 100 times greater than air.

Because of the difficulty of predicting analytically the performance of cold plates, it is usually necessary to construct several experimental plates in order to determine the exact design parameters. In one instance (Ref. 18), a cold plate which was fabricated by milling slots in a thick aluminum plate and screwing a cover over the top to form the cooling passages was found to be unsatisfactory. Subsequent cooling plates were made with plate-and-fin-type heat transfer surfaces of dip-brazed aluminum with inlet and outlet headers. A variety of internal arrangement types were constructed with different fin configurations. A heat dissipation of 1.0 watt/sq.in. of cold plate surface area was obtained, using air at sea level pressure. Both sides of the cold plate were used. Hence, a 12" x 12" cold plate could dissipate 12 x 12 x 2 = 288 watts total. In order to minimize the temperature drop between the electronic packages and the cold plate, clamps were required to provide adequate contact pressure. The major difficulty encountered during this development was in maintaining the surface flatness of the cold plates. See Fig. 37.

While most of the current literature deals with cold plates for high altitude airborne electronic equipment, the basic theory and design techniques to be used for sea level operated forced air cooled cold plates and for liquid cooled cold plates are similar.

In general, there are three major thermal resistances to be considered in cold plate design. First, there is the resistance from the heat producing part to the metal wall of the case. This may be the greatest and, perhaps, the controlling resistance. It can be minimized by



Courtesy RAYTHEON MFG CO

CROSS SECTION THROUGH A TYPICAL FORCED AIR COOLED COLD PLATE

Fig 37

filling the container with a better heat transfer medium, such as silicone fluid, or by providing conduction paths of metal from the surface of the part to the container walls or by a combination of the two. The second major resistance is that formed at the junction of the container and the cold plate. This can be minimized by intimate metal-to-metal contact under pressure. The third resistance is that offered by the coolant flowing in the cold plate heat exchanger. This may be relatively high. If air is the fluid, its flow must be turbulent to be at all effective in reducing this high resistance. On the other hand, if fresh water is the coolant, the resistance would be made relatively very low, in which case the first resistance (part-to-case) would probably control the rate of heat transfer.

Care must be used in determining the pressure drop in air cooled cold plates. Several of the current techniques for measuring pressure gradients appear to be questionable. The pressures should be measured in the main ducts at points of laminar flow not in headers and nozzles where pressure probes will indicate a range of values dependent upon their location.

VII. LIQUID COOLING

A. GENERAL

Liquid cooling systems are applicable for use with electronic equipments which are to be designed for operation in thermal environments of high ambient temperature or with high heat dissipations. In general, liquid cooling can be four or five times as effective as forced air cooling. Liquid cooling systems may be basically classified as either direct or indirect. In a direct system the coolant is in direct contact with the electronic parts. Heat is conveyed directly from the heat producing parts to the coolant, which serves as the vehicle for transferring the heat. In this case, the liquid is the primary mode of heat removal from the parts. In an indirect system, the liquid coolant does not come in direct contact with the electronic parts. Heat is removed from the parts by natural convection, conduction, radiation or forced convection to a liquid cooled panel or heat exchanger.

Liquid coolant systems may be divided into five types which, listed in their approximate order of complexity, are:

1. Direct liquid immersion
2. Direct liquid immersion with forced circulation
3. Indirect liquid cooling
4. Direct spray cooling
5. Composite indirect liquid cooling systems

1. Shipboard Applications of Liquid Cooling

In recent years the miniaturization of equipment and increased power dissipations have increased the heat concentrations of electronic equipment within enclosures to the point that the limit of shipboard air conditioning and ventilation systems has sometimes been exceeded. In addition to exceeding the thermal limits of the electronic components, the high heat concentrations have made small rooms or cabins almost uninhabitable for personnel. With continuous liquid cooling systems, electronic equipments in enclosed spaces may be cooled without adding heat to the compartments. This may simplify the ventilation problem. Perhaps the most practical form of liquid cooling for miniaturized shipboard equipment is the cold plate heat exchanger to which subassemblies may be attached. The subassemblies should be designed for cooling by natural means with a path of low thermal resistance to their base for transferring the rejected heat into the cold plate. Alternately, the subassemblies may be forced air cooled to transfer their heat into an air to liquid cold panel heat exchanger.

2. Coolants

There are a number of volatile liquids that can be generally classed as liquid coolants. However, this section pertains only those liquids which are highly stable and non-volatile at the selected operating temperature and do not undergo a change in state.

3. Heat Exchangers

The design of heat exchangers has not been emphasized in this Manual. The heat exchangers which are discussed in this section are based on the design assumption that, on shipboard, fresh cooling water is available at 35°C and in sufficient quantities that the secondary coolant in the heat exchanger will have about a 3°C temperature rise. The temperature of the coolant leaving the heat exchanger should not be lower than about 38°C. If the coolant is at a lower temperature, the piping may be below the dew point temperature of the ambient air, and condensation of atmospheric moisture may take place, being apparent in extreme cases as "sweating" of the pipes and associated electronic equipment. Condensation must be avoided, especially when electronic equipment is located in compartments having air of high humidity. Because of this condition the higher coolant temperatures (38°C) have been selected for use in design examples. For operation in the tropics, it is recommended that 50°C coolant temperature be used in order to prevent condensation damage aboard ship, in vehicles and in all ground based equipment.

B. THEORY

1. General

As previously mentioned, electronic parts normally dissipate heat by radiation, free convection, metallic conduction and, where the spaces between parts are small, by gaseous conduction. When the surrounding fluid is a liquid, the effects of radiation dissappear and the heat is dissipated by combined free convection and conduction. The gaseous conduction effect is present, particularly in densely packaged units. In free air considerable heat transfer between short vertical surfaces occurs by gaseous conduction for distances up to about 1/4 inch. It is probable that a similar situation exists when a liquid, rather than air, is the surrounding fluid. Thus, both high thermal conductivities and convection coefficients are desirable.

The equations for free convection from unconfined surfaces given in Section V are applicable to any fluid, gaseous or liquid. Unlike air, however, simplified charts are not available for free convection in liquids and equations must be used in estimating the film coefficients. The general form of the equation for free convection from an unconfined surface is also given in Section V.

A general discussion of the heat transfer from an electronic part immersed in a liquid is in order. Consider a heat producing part, say a vacuum tube within a small liquid filled metal box which (for simplicity) is suspended in air. The heat must be transferred first by free convection from the surface of the tube to the liquid, then by free convection from the liquid to the walls of the box and then by conduction through the metal walls. From the walls, the heat is transferred to the surrounding air by combined convection and radiation. The thermal circuit is best visualized by an electrical analogy wherein each thermal resistance is represented by an electrical resistance. The heat transfer is represented by the current and the overall Δt (surface temperature of part minus surrounding air temperature) and by the voltage required for the flow of the current through these resistances. Fig. 38 shows the equivalent electrical circuit.

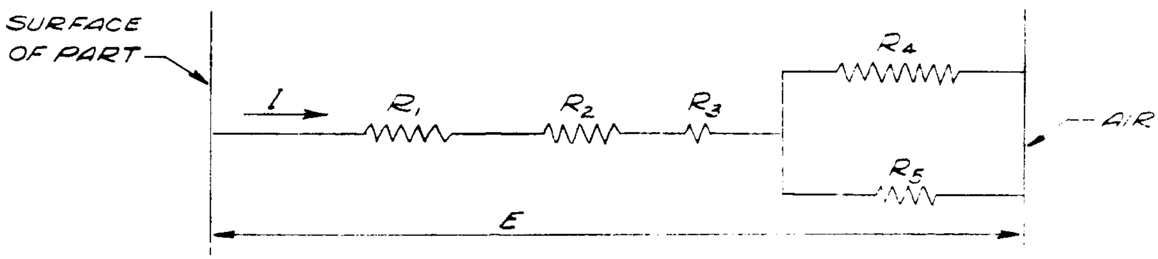


Fig. 38 Equivalent Electrical Circuit

Electrical Resistance	Corresponding Thermal Resistance
R_1	Liquid film on surface of parts
R_2	Liquid film on inner surface of box
R_3	Walls of box
R_4	Air film on outer box surface
R_5	That due to radiation

The current flow is given by:

$$I = \frac{E}{R_1 + R_2 + R_3 + \dfrac{R_4 R_5}{R_4 + R_5}}$$

It may be difficult to solve such thermal circuits since the resistances are not well defined, hence, only approximations are normally utilized.

The analogy is useful in determining the relative value of the Δt's across each resistance. In the example discussed here, the resistance due to conduction across the metal walls of the box is probably small, in fact, almost negligible compared with the parallel resistance combination of R_4 and R_5 due to free convection in air and radiation. Since the resistance due to free convection in liquids is considerably less than that for air, R_1 and R_2 each will be considerably smaller than the R_4 and R_5 combination. Thus, the main resistances in this instance will be caused by the free air convection and radiation combination with a relatively smaller contribution to the overall R due to R_1 and R_2, the R_3 (metal walls) being negligible. The Δt's across each resistance will correspond to the relative values of the resistances. This "relative resistance concept" should always be kept foremost in mind. In the liquid filled box example, a given q or wattage dissipation may cause the surface temperature of the part to become excessive, due to the poor heat transfer properties of the free air outside the box, in spite of the much better heat transfer properties of the liquid. It is important to consider the overall heat transfer problem and the effect of a high series resistance far removed from the part in question, on the surface temperature of that part.

2. Example 6

To illustrate the use of heat transfer calculations, the following example approximating the average surface temperature of an electronic part immersed in a liquid is presented:

Given:

A thin metal case 1 1/2" x 3/4" x 3 1/2" high is painted black on the outside and suspended in free air at 100°C. All surfaces which the case "sees" are assumed to be at 100°C.

Three subminiature vacuum tubes, each approximately 1 3/4" high by 3/8" dia. (bulb surface area of 2.18 sq. in.) are within the case which is filled with silicone fluid (550-DC, 112 centistoke grade). Each tube is to dissipate 3.5 watts and it is assumed that conduction through all lead wires is negligible.

Problem:

Approximate the average bulb surface and hot-spot temperatures.

Solution:

First step: Determine the average outer case temperature.

q = 3 x 3.5 watts x 3.413 Btu./hr./watt = 35.8 Btu./hr.
Area of case, A = 18 sq. in. = 0.125 sq. ft.
The 35.8 Btu./hr. must be dissipated from the outer surface of the case by combined radiation and free convection. The determination of this temperature is made by a trial and error solution:

q by radiation: $q_r = 0.173A \, F_e \, F_a \left[\left(\frac{T_s}{100} \right)^4 - \left(\frac{T_a}{100} \right)^4 \right]$ (28)

q by free convection: $q_c = h_c A \, (t_s - t_a)$ (3)

The value of T_s must be determined such that $q_r + q_c = 35.8$ Btu./hr.

Radiation:

A = 18 sq.in. = .125 sq. ft.
F_a = 1.0 (assuming case situated in relatively large space)
F_e = emissivity outer case = 0.85
t_a = $100^oC = 212^oF$
T_a = $460 + 212 = 672^oR$
T_s = $t_s + 460$

Thus $q_r = 0.173 \times 0.125 \times 0.85 \times 1.0 \left[\left(\frac{T_s}{100} \right)^4 - \left(\frac{672}{100} \right)^4 \right]$

$$q_r = 0.0184 \left[\left(\frac{T_s}{100} \right)^4 - 2045 \right] \qquad (50)$$

Convection: The film coefficient for a vertical surface differs from that for a horizontal surface facing upwards; likewise, for a surface facing downward. To simplify this part of the problem, it is assumed that the film coefficient for a vertical surface is the average over the entire surface. The equation for the film coefficient for a vertical surface is:

$h_c = 0.55 \frac{k}{L} (a \, L^3 \Delta t)^{1/4}$ (19)
k = thermal conductivity
L = height of case = $\frac{3.5}{12}$ = 0.292 ft.
$\Delta t = t_s - t_a = t_s - 212^oF$.
a = free convection modulus
Substituting (19) in (3), using the above values:

$$q_c = 0.55 \, \frac{k}{0.292} \, (a \times 0.292^3 \Delta t)^{1/4} \, (0.125) \Delta t \qquad (51)$$

or

$$q_c = 0.0937 \, k \, a^{1/4} \Delta t^{5/4} \qquad (52)$$

Thus,

$$q_t = q_r + q_c = 0.0184\left[\left(\frac{T_s}{100}\right)^4 - 2045\right] + 0.0937\ k(a)^{1/4}\Delta t^{5/4} \qquad (53)$$

$$= 35.3\ \text{Btu/hr.}$$

Assume:

$\Delta t = 70^{\circ}F$

$t_s = 212 + 70 = 282^{\circ}F$

$T_s = 460 + 282 = 742^{\circ}R$

Av. film t = $(282 + 212)/2 = 247^{\circ}F$

k @ $247^{\circ}F = 0.0192$ Btu./(hr.)(ft.)($^{\circ}F$)

a @ $247^{\circ}F = 0.425 \times 10^6$

$q_r = 0.0184\ [(7.42)^4 - 2045] = 18.3$ Btu./hr.

$q_c = .0937 \times 0.0192 \times (0.425 \times 10^6)^{1/4} \times 70^{5/4} = \underline{9.3}$ Btu./hr.

Total $= 27.6$ Btu./hr.

It is seen that a surface temperature of $282^{\circ}F$ allows only 27.6 Btu./hr. to be dissipated from the case. Therefore, a higher surface temperature must be assumed.

Assume:

$\Delta t = 90^{\circ}F$	av. film t $= 257^{\circ}F$
$t_s = 302^{\circ}F$	k @ $257^{\circ}F = .0194$
$T_s = 762^{\circ}R$	a @ $257^{\circ}F = 0.40 \times 10^6$

By similar calculation:

$q_r = 24.6$ Btu/hr $\qquad q_c = 12.7$ Btu/hr.

$q_{total} = 37.3$ Btu/hr.

Interpolating at 35.8 Btu/hr.:

$\Delta t = 86.9^{\circ}F$

and $\quad t_s = 212 + 86.9 = 299^{\circ}F$

<u>Second Step</u>: Conduction through metal walls. Assume metal walls are of steel 0.0375" thick:

k for steel = 33 Btu/(hr.)(ft.)($^{\circ}F$)

Then, since $q = \dfrac{\Delta tA}{\frac{L}{k}}$ (54)

$$t = 35.8 \times \frac{0.0375}{12 \times 35} \times \frac{1}{0.125} = 0.027^{\circ}F$$

Thus, the steel walls offer a negligible resistance to heat transfer and can be completely disregarded.

Third Step: Determine the silicone fluid temperature. The heat transfer between the silicone fluid and the metal walls is assumed to be by free convection, although the problem is complicated due to the confinement of the liquid. Again equation (52) is used to estimate the Δt. Since the silicone fluid is better than air in free convection, the Δt between the silicone and walls will be considerably lower.

Assume:

$$\Delta t = 20^{\circ}F$$

$$t_{silicone} = 299 + 20 = 319^{\circ}F$$

av. film temp. = 299 + 10 = 311°F

k @ 311°F = 0.0703 Btu/(hr.)(ft.)(°F)

a @ 311°F = 2.33×10^{8}

A = 0.125 sq. ft.

L = 0.292 ft.

Thus:

$$q = 0.0937 \times .0703 \times (2.33 \times 10^{8})^{1/4}(20)^{5/4}$$

$$q = 34.5 \text{ Btu/hr.}$$

This is close to the required 35.8 Btu/hr. so

$$\Delta t \text{ (approx.)} = \frac{35.8}{34.5} \times 20 = 20.8, \text{ say } 21^{\circ}F$$

Thus, silicone temp. = 21 + 299 = 320°F

Fourth Step: Determine average envelope temperature. Again, the heat transfer process is assumed to be by free convection in spite of confinement and dense packaging. Since, for small electronic parts, the values of C and m are equal to 1.45 and 0.23 respectively, the equation becomes: (See Section V).

$$q = h_c A \Delta t = 1.45 \frac{k}{L} \left(aL^3 \Delta t\right)^{.23} A \Delta t \quad (21)(22)$$

$$\text{or} \quad q = 1.45 \frac{k}{L} (aL^3)^{.23} (\Delta t)^{1.23} A$$

where:

$$L \text{ for tube} = \text{height} = \frac{1.75}{12} = 0.146 \text{ ft.}$$

$$A = 3 \times 2.18/144 = 0.0454 \text{ sq. ft.}$$

Thus,

$$q = 0.12k(a)^{0.23}(\Delta t)^{1.23}$$

Try: $\Delta t = 30^{\circ}F$

$$t_{envelope} = 320 + 30 = 350^{\circ}F$$

$$\text{av.film } t = 320 + 15 = 335^{\circ}F$$

k @ $335^{\circ}F = 0.0695$
a @ $335^{\circ}F = 2.65 \times 10^{8}$
$q = 0.12 \times 0.0695 \times (2.65 \times 10^{8})^{.23} \times (30)^{1.23}$
$q = 47.6$ Btu/hr (this is too high)

Try:

$$\Delta t = 24^{\circ}$$
$$t_{envelope} = 320 + 24 = 344^{\circ}F$$
$$\text{av.film } t = 320 + 12 = 332^{\circ}F$$
$$k \text{ @ } 332^{\circ}F = 0.0696$$
$$a \text{ @ } 332^{\circ}F = 2.61 \times 10^{8}$$
$$q = 0.12 \times 0.0696(2.61 \times 10^{8})^{0.23}(24)^{1.23}$$
$$q = 36.1 \text{ Btu/hr.}$$

This is close to the required 35.8 Btu/hr., so Δt is approximately $24^{\circ}F$. Thus, the average surface temperature of tube envelope is $320 + 24 = 344^{\circ}F$ or $173^{\circ}C$.

It must be noted that the $173^{\circ}C$ is an average temperature and does not represent the hot spot temperature. The ratio of maximum temperature rise (hot spot minus fluid temperature) to average temperature rise varies with the shape and type of tube and method of cooling. For free convection, with subminiature tubes, the maximum temperature rise occurs slightly above the midpoint and is about 15 per cent greater than the average temperature rise. For miniature tubes, it is about 22 per cent. Thus, in the foregoing example, the maximum temperature rise is:

$$1.15 \times 24^{\circ}F = \text{approx. } 28^{\circ}F$$

$$\text{Maximum tube temp.} = 28 + 320 = 348^{\circ}F \text{ or } 176^{\circ}C$$

The $176^{\circ}C$ maximum bulb temperature must be considered only as a fair approximation, due to the many assumptions and complexities involved.

3. <u>Comparison of Fluids in Free Convection</u>

An interesting comparison of several fluids in free convection, based on experimental work, is given in reference (10). This comparison is given in Table 10. To understand the table, the experimental set-up and conditions must be explained.

The test apparatus consisted of a closed cylindrical container in which seven #5763 miniature tubes were located, all equidistant from each other; i.e., six tubes surrounding a middle or center tube, all tubes in the vertical position. The container was immersed in a controlled temperature oil bath. Four fluids were tested; air at sea level pressure, silicone fluid, transformer oil, and Freon-113. In all tests, the fluid temperature was maintained constant as was the heat dissipation from the center tube and total heat dissipation from the test container. The container wall temperature was varied to maintain this total dissipation for the constant fluid temperature.

TABLE 10

Comparative Heat Transfer Data for Air, Silicone Fluid, Transformer Oil, and Freon in Multi-Tube Apparatus

Fluid temperature	130°C
Unit heat dissipation	3 watts/sq.in.of tube envelope
Total heat dissipation from enclosure	115 watts

	Tube Surface			Enclosure Wall	
Fluid	Temp. Rise Over Fluid °C	Temp. °C	Surface Heat Transfer Coefficient Watts/(sq.in)(°C)	Temp.Drop (Below Fluid) °C	Temp. °C
Air (Sea Level)	105	235	0.029	148	-18
Silicone Fluid	19.1	149.1	0.157	31	99
Transformer Oil	13.5	143.5	0.222	19	111
Freon - 113	5.6	135.6	0.535	7	123

Table 10 shows the great advantage obtained in using a liquid rather than air as a medium of free convection. In the experiments the enclosure walls had to be maintained at -18°C using air as the fluid whereas, for transformer oil, the wall temperature could be raised to 111°C for the same fluid temperature and heat dissipation. Also, the tube surface temperature for air was 235°C and it was only 143.5°C for the transformer oil.

C. DIRECT LIQUID IMMERSION

1. General

A simple direct liquid cooling system is shown in Figure 39. The electronic parts of the subassembly are completely immersed in a fluid, such as DC-200 Silicone fluid. The heat is transferred

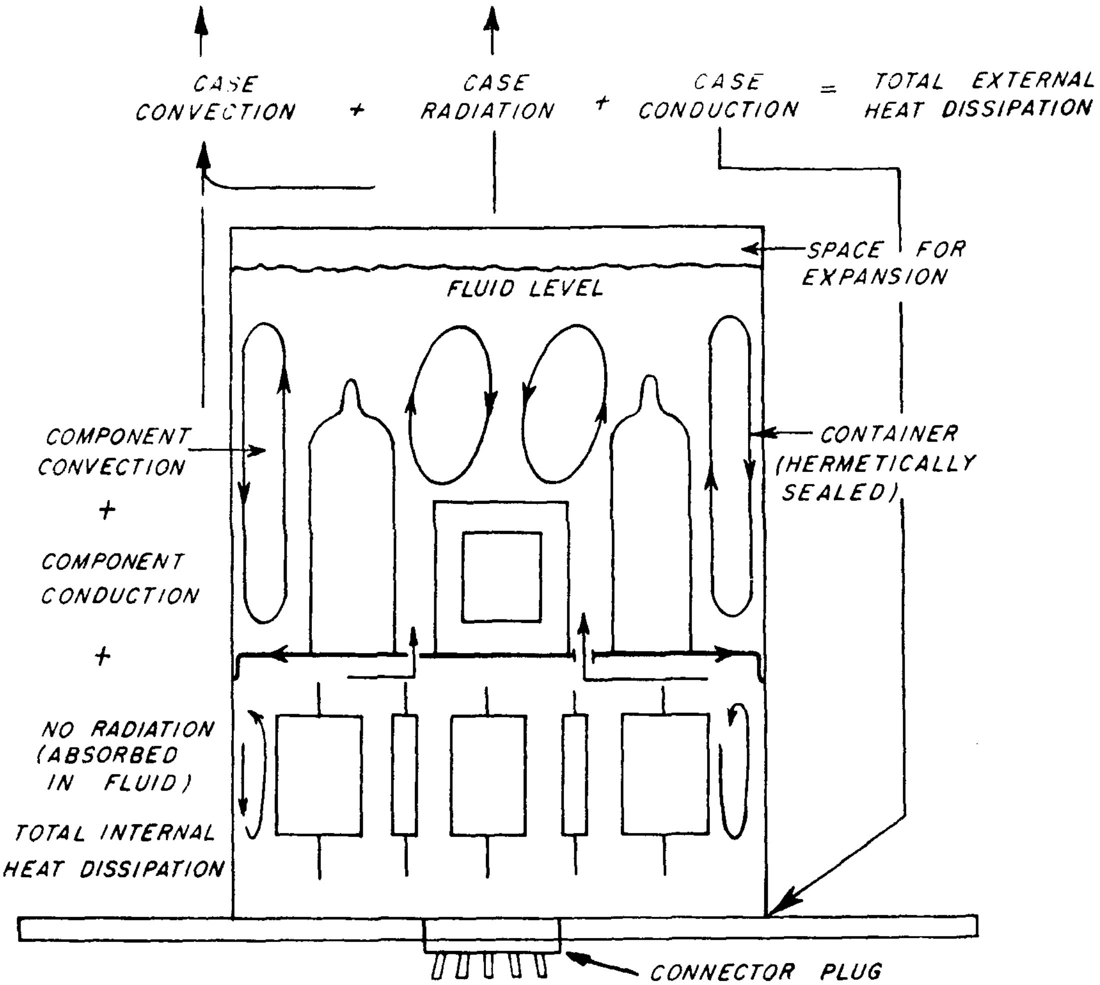


Fig. 39

HEAT TRANSFER DIAGRAM OF A SIMPLE DIRECT LIQUID COOLED SUBASSEMBLY from the parts to the surfaces of the container by free convection and molecular conduction in the fluid. Radiation from the heat producing parts is nil. This type of subassembly may be designed to dissipate a maximum of the order of one half watt per square inch of effective surface area in free air and as much as two or three watts per cubic inch when more effective external cooling is applied.

Free convection currents are produced in the fluid in the following manner: Upon contact with the hotter surface of the electronic parts, the coolant fluid expands, reducing its density. The coolant rises until it comes in contact with the colder surface of the container where the heat is removed. The fluid contracts, increasing its density, and falls. In free convective cooling, the temperature distribution is not uniform, the temperatures are low at the bottom of the container and gradually increase to the highest values near the top.

2. Design Considerations

The design problems encountered in a liquid immersed subassembly include hermetic sealing, provision for expansion of the liquid coolant, vapor pressure, strength of the container, ease of repair, removal of the coolant from the parts, orientation of parts and the effects of the coolant on the function of high frequency circuits. The orientation and mounting of parts must be given special consideration since they must be located so as to achieve maximum convection. Metallic conduction paths of low thermal resistance from the heat producing parts to the surface of the case are not normally necessary.

a. Hermetic Sealing

Oxidation of coolants may be minimized by hermetic sealing of containers of electronic equipment. One technique in use involves filling and sealing the container with heated coolant while the entire assembly is heated to a temperature that is higher than the peak anticipated operating temperature. When the fluid cools, the vacuum-like space of volatile constituents above the fluid allows for expansion of the fluid. The container must be strong so that it cannot collapse inwardly. The fluid can completely fill the container provided that the operating temperature is never permitted to exceed the filling temperature. Otherwise, the increased internal pressure may cause structural failure of the case. It is recommended that units that are subjected to changes in altitude, such as those used for mobile or airborne equipment, be hermetically sealed. Induction soldering with high temperature solder may be used to seal off the component package. Fluid expansion may also be compensated for by a rubber diaphragm, a metal bellows arrangement, or a rubber ball mounted inside of the case. If electronic equipment is to be operated in a fixed location and always remain in an upright position, it may be possible to operate the electronic assembly in a container of fluid which is vented to the atmosphere. Further, if desired, non-spillable vents such as are used on aircraft batteries may be provided. Some of the difficulties encountered with fluids that have large coefficients of expansions can thus be avoided.

b. Mechanical Considerations in the Mounting of Electronic Parts

It is necessary to expose a maximum of the surface of the heat producing parts to the coolant and to direct the free convection currents of the fluid around these parts. Thus, in liquids, parts should be mounted, as in air, to promote convective cooling (See Section VI). Metallic conduction paths of low thermal resistance from the heat producing parts to the surface of the case are not as important in direct liquid cooled subassemblies as in most other types of subassemblies, since adequate cooling is usually obtained by convection. The parts may be supported by insulators or insulating materials so long as the fluid is permitted to freely circulate around the parts.

The construction of this type of equipment must be given special consideration. With viscous coolants, a slight mechanical advantage is gained by the immersion of the electronic parts in the fluid, since the fluid tends to lend support to tubes and other parts. Also, it can provide a damping action which assists in resisting vibration and shock, dependent upon the viscosity of the fluid. Subminiature tubes can be mounted by their lead wires and can be supported at their other end by a loop of wire around the tip of the tubes. Thus, almost the entire tube envelope is exposed to the coolant fluid. The tubes should be mounted vertically with their bases downward. This provides maximum cooling at the hot spots and tends to minimize electrolysis. Holes can be provided in the subchassis to direct the flow of the free convection currents around the other heat producing parts.

c. Coolant Selection

When selecting a coolant, it is necessary to consider the change in the thermal and physical characteristics of the liquid over the entire operating temperature range as well as its chemical and electrical compatibility with the metals and materials with which it comes in contact. With direct liquid cooling systems, these properties include chemical inertness, dielectric constant, power factor, viscosity, vaporization temperature, freezing temperature, flash point, vapor pressure, toxicity, coolant life, thermal coefficient of expansion, thermal coefficient of viscosity, surface tension, thermal conductivity and dielectric strength.

There are a number of coolants which have been used in subassemblies, power and communications transformers, chokes. capacitors, and other high voltage or high temperature parts. These fluids are for the most part hydrocarbons of one form or another and suffer from deficiencies, such as high dielectric constant, molecular instability, high power factor (particularly at radio frequencies), and inability to withstand very low or very high temperatures.

Silicone fluids are superior to the hydrocarbons, especially in regard to their ability to operate at temperature extremes. High temperature operation is limited only by the cracking temperature of the fluid. The silicones are available in a wide range of viscosities and the electrical characteristics of several are listed in this section. The dielectric constant and power factor are good up to at least 100 MC. The dielectric constant decreases slightly while the power factor increases with increasing temperature. In most instances these changes will be insignificant. For example, 500 centistoke silicone fluid changes power factor from .000025 to .0003 in going from 25°C to 150°C at a given frequency.

The thermal conductivity of silicone fluids is actually quite low, being intermediate between glass and air. For this reason, a rather low viscosity fluid is generally chosen so that circulation due to convection will be aided. Fifty centistoke silicone fluid has been used with considerable success.

The thermal expansion coefficient of all liquid dielectric materials is high. For example, DC-200 silicone fluid increases in volume by 13 percent over a temperature range of from 25°C to 150°C. It is therefore necessary to provide adequate space for expansion.

It has been reported that polystyrene and some of the related resins are attacked by chlorinated hydrocarbons. Certain organic paints and varnishes are attacked by transformer oils. It has also been found that silicone fluids have an undesirable effect on some silicone protected parts. In general, however, silicone fluids are inert with respect to most commonly used electronic materials. Data can be obtained from the manufacturer.

Freons have not been utilized for direct liquid cooling because of the resultant high operating pressures at temperatures of the order of 100°C.

Petroleum base oils can be used in liquid cooled equipments whose "hot spot" temperatures do not exceed 175°C. The oils oxidize or rapidly decompose at higher temperatures and must be changed occasionally. Mineral oil exhibits lower electrical losses at high frequencies than the silicone fluids.

Impurities cannot be tolerated in coolants used in liquid filled subassemblies. Even a small percentage of moisture can lead to electrolysis and rapid corrosion of wires and leads, especially those with DC circuits in excess of 25 volts. Extreme oare must be used to prevent contamination of the coolant. The physical characteristics of coolants are presented later in this section and in Appendix C.

d. Temperature Distribution Around Liquid Cooled Subminiature Tubes

A type 6K4 tube, supported by its leads, was **immersed** in "chemically pure" ethyl alcohol, in a **transparent** l.... box. **After** plate voltage was applied, there was considerable electrolysis around the tube leads due to the moisture and impurity content of the alcohol. When the tube was mounted in the horizontal position and illuminated by a point source of light, interfaces between parts of the alcohol at different temperatures could readily be distinguished. The areas within the liquid where heating occurred showed up clearly. See Fig. 40.

The area at A is a convection current of strongly heated liquid rising upward to the surface of the alcohol. The area at B is also a convection current of heated liquid which is rising more slowly to the surface. The area at C is an area of apparently stationary liquid which is heated by radiation directly from the surface of the glass envelope. A distinct feature of this radiation area is that, regardless of tube positioning, it is always the same. The shape of the other areas depend directly on positioning. The radiation area around the tube is directly opposite the plate of the tube and the constriction occurs between the plate and the seal of the tube. The thickness of this halo-like light zone indicates relative amounts of heat radiated into the surrounding liquid and is apparently due to a change in the index of refraction of the coolant. The zone appeared and disappeared very slowly as plate voltage was switched on or off and was thus shown to be entirely dependent on tube plate temperature.

Observations

During the test several electrolysis effects were noted. These produced considerable heat around the terminals of the plastic box and around the tube leads themselves. It was found that, due to this cause, heating effects on the tube leads could not be determined. The positive tube lead soon turned black and other tube leads were heavily coated with cupric salts.

D. DIRECT LIQUID IMMERSION WITH AGITATION

It was previously stated that free convection in liquid coolants did not provide an even temperature distribution within the container. By the addition of a small motor driven agitator, relatively constant liquid temperature can be maintained (see Fig. 41). For example, it was found that the temperature difference within the fluid was reduced to 0.6°C, using a five vane agitator when the wall temperature gradients were less than 5.5°C. A three vane agitator in a six inch diameter container of silicone fluid maintained an average temperature difference of 2°C within the container at 660 rpm, 1°C at 1440 rpm. With free convection liquid cooling and a unit heat dissipation of 0.1294 watts per square inch surface area, the temperature rise between the bottom and top of the container was 10°C. With a unit heat dissipation of 1.06 watts per square inch and 50 centistoke DC-200 Silicone fluid, the temperature rise was in the neighborhood of 46°C, (ref.10).

These figures are for package sizes that are normally larger than those used for miniaturized equipment. The same degree of cooling cannot be expected in miniature packages. However, these tests indicate that certain configurations can be more effectively cooled by expending a small amount of power for agitation of the coolant.

E. DIRECT FORCED LIQUID COOLING

1. General

Just as increased cooling is obtained with forced air instead of natural air convection, forced circulation of the coolant greatly increases the cooling rate when liquid cooling is used. The coolant may be pumped into an external exchanger for transfer of the heat into the sink. The pump, of course, requires power for its operation and, unfortunately, most of this energy is expended in the coolant in the form of heat. Thus, the total heat rejected is increased by the power consumed by the pump. However, the advantages of the small temperature gradients and increased cooling rates achieved with forced liquid cooling can easily outweigh the disadvantages of the increased system complexity.

2. Direct Forced Liquid Cooling

A liquid cooling system of this type is presented by Fig. 42. The electronic components are completely immersed in a compatible liquid coolant such as silicone fluid, transformer oil, or freon. A low-pressure pump circulates the coolant liquid through the system. The accumulator (air cushion tank) allows for expansion of the fluid and serves to minimize vapor lock in the system. The heat exchanger removes the heat from the liquid before it is recirculated through the electronic equip-

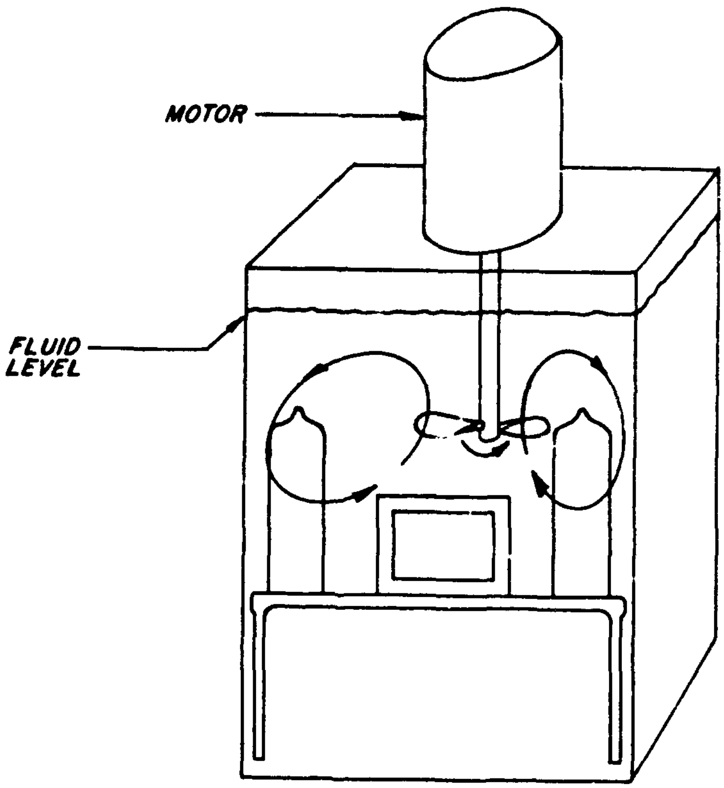


Fig. 41

DIRECT LIQUID COOLING WITH AGITATION

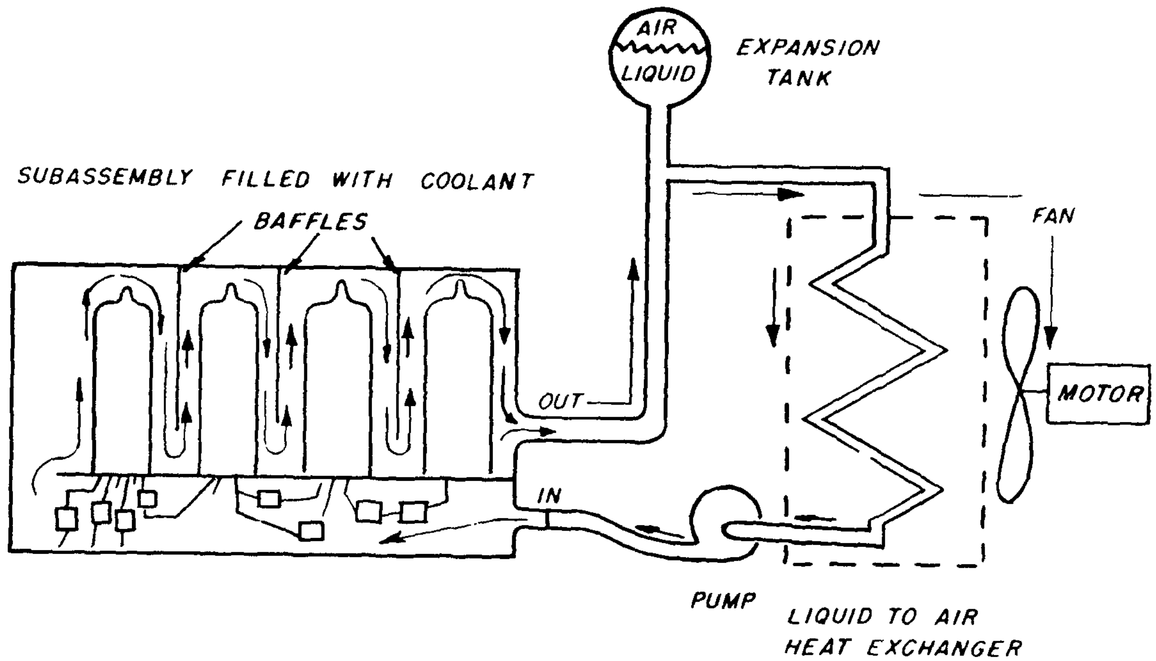


Fig. 42 DIRECT FORCED LIQUID COOLING SYSTEM

ment. Care must be exercised in orienting parts in order to obtain maximum cooling effectiveness. Further, if relatively high pumping pressures are used, the high pressure stream should not be directed upon fragile electronic parts. The mathematical treatment of forced liquid cooling design is identical with that used for forced air cooling other than that different coefficient values are used.

3. Direct Spray Cooling

Fig. 43 shows a direct spray-cooling system for miniaturized electronic equipment. The coolant liquid, under a slight pressure, is pumped into a manifold for distribution to spray nozzles that are located so as to cover the electronic part to be cooled with a continuous film of liquid. Thus, the sprayed liquid absorbs heat from the electronic parts by direct contact. The heated coolant is then collected in a sump in the bottom of the equipment and pumped through a heat exchanger to be cooled before it is returned to the manifold for re-distribution. With spray cooling, it is possible to adjust the number of jets and the volume of flow to obtain optimum cooling of each individual heat producing part, whereas, in some of the direct liquid cooled systems there is a tendency either to undercool or overcool individual electronic parts. Further, the amount of coolant used in a spray system may be much less than that of a liquid filled system.

Spray-cooling is most applicable to parts such as vacuum tubes when they are mounted in the vertical position. However, satisfactory results have been obtained with tubes in the horizontal position by covering part of their envelopes with tight-fitting copper screen sleeves. These sleeves will provide more even distribution of the fluid around the envelope hot spots.

One of the difficulties encountered in spray-cooling systems is to obtain uniform distribution of the fluid over the part to be cooled. This depends upon the wetting qualities of the coolant, the number, size and placement of the spray jets, and the rate of flow of the coolant through the jet nozzle. Changes in the rate of flow of the jets modifies the flow pattern of the coolant and thus the rate of cooling of the electronic parts. In a reasonably efficient spray cooling system, the additional power required for pump operation will be of the order of five percent of the total dissipated power after a certain critical dissipation has been reached. Below this minimum dissipation (imposed by available pump sizes) the percentage of dissipated power required by the pump may be greater than 5%. An efficient spray cooling system should require only about one third of the amount of liquid required by an equivalent liquid filled unit.

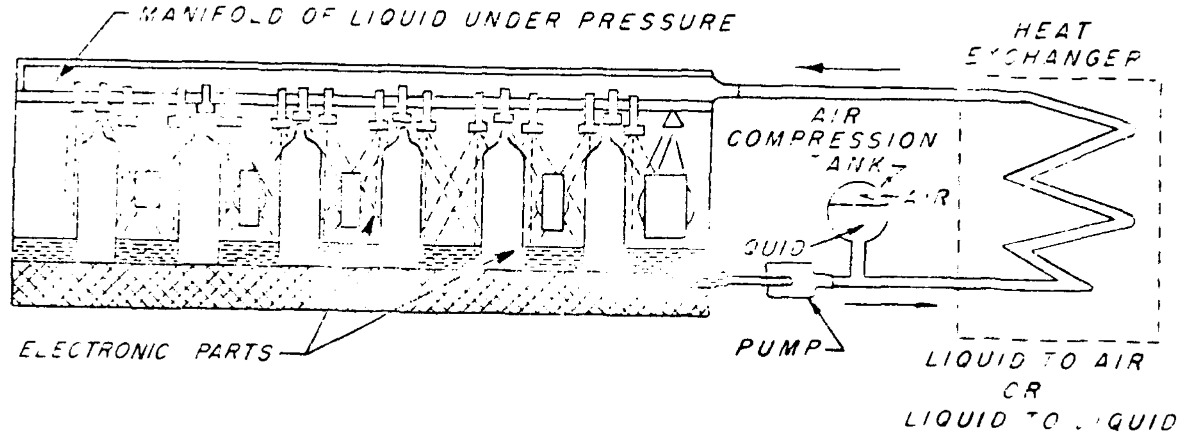


Fig 43 – DIRECT SPRAY COOLING SYSTEM

a. Nozzles

Extremely small nozzle bores are usually necessary in order to obtain a fine spray. In conventional electronic equipment the jet spray has proven most satisfactory because of its wide spray angle which permits a compact equipment design. Further, a jet spray nozzle emits a solid stream of liquid a short distance, it does not require a high pressure pump and it is not as susceptible to plugging, since the bore is relatively large.

A starting point for jet size in experimental development at low spray rates is a number 80 hole drilled in a #2 screw. The screw may be inserted into a hole tapped into the manifold and adjusted to the proper height and angle above the parts to be cooled. This provides an easy method of experimenting with the number and the size of the jets necessary to cool a given equipment.

b. Nozzle Manifold

The spray distribution manifold can be designed to be an integral part of the unit's housing or as an independent part. In either system the initial model should be made in such a manner that the manifold is accessible for changing the size, number and placement of jets until satisfactory cooling of the subassemblies is obtained.

4. Detailed Design Considerations

a. Pump Selection

Either gear reciprocating or centrifugal type pumps may be used for silicone and similar liquids. However, it is usually preferable to use gear type pumps, since they do not become airbound as readily as centrifugal units. Further, gear pumps have the additional advantages of smaller weight and size and a higher operating efficiency.

The centrifugal pump has an advantage for this application in that is does not have a constant displacement and the volume of flow may be controlled by a valve or variable orifice in the discharge line without overloading the motor. Less power will be required when pumping smaller quantities of fluid. Also, a rheostat in series with the centrifugal pump motor may be used to simultaneously adjust the rate of flow and the pressure. Unfortunately, centrifugal pumps become airbound and must be primed prior to operation. This can be overcome by mounting the pump below the liquid level so that the pump inlet is always flooded.

Positive pressure rotary vane and reciprocating pumps are usually self-priming and suitable for handling non-lubricating liquids. However, because of the constant displacement, the throttling of such pumps increases motor load and pressure but does not decrease the coolant flow.

For the selection of a pump for a given cooling system, it is necessary to consider the following:

(1) The physical and chemical characteristics of the coolant liquid being used, such as specific gravity, viscosity, temperature, solids in suspension, abrasive material, thermal stability, and the corrosive and solubility effects on materials used in pump construction.

(2) The state of the fluid at the pump inlet, whether it floods the inlet, is full of air bubbles, whether the pump must prime itself and, if so, the priming lift.

(3) The pump characteristics desired in terms of delivery required, differential pressure, inlet pressure, discharge pressure, the sum of the differential pressures across the electronic subassemblies, the heat exchanger and the line loss, and the duty cycle.

(4) The voltage, phase and frequency desired for the pump motor and environmental conditions, such as dust, moisture, fumes and fire hazards under which it must operate.

(5) The electrical and mechanical noise characteristics of the pump motor.

b. Air Cushion Tank

The air cushion tank is provided to allow for expansion of the fluid as its temperature increases, to remove air from the coolant, and to cushion the shock in the entire system if the pump should become vapor bound. The tank should be large enough to allow for the expansion of all of the liquid in the system and still provide an air cushion.

c. Heat Exchanger

The heat exchanger should be of a size adequate to remove the rejected heat from the coolant fluid before the fluid is re-circulated through the electronic equipment. A liquid to forced air heat exchanger, similar to an automotive radiator could be used for ground based equipment.

For most shipboard applications a liquid to liquid heat exchanger would probably prove most practical, since an adequate supply of cooling water is usually available.

In selecting a heat exchanger for a given system, it is necessary to determine the following:

(1) What secondary cooling means or "sink connection" is available.

(2) The rate of flow of coolant fluid through the heat exchanger.

(3) The pressure drop in the coolant fluid across the heat exchanger.

(4) The temperature of the coolant fluid when it enters and when it leaves the heat exchanger.

(5) The expected limits of the thermal environment in which the heat exchanger must function.

d. Design Notes

(1) General

Properly applied, direct liquid immersion can be an effective cooling method. Subassemblies should be constructed so that the heat producing parts are separated from the non-heat producing parts, especially those that are temperature sensitive. The coolant arriving from the heat exchanger should be directed onto the temperature sensitive parts first and later be directed to the heat producing parts.

Heat transfer inside a liquid cooled electronic assembly may be difficult to predict due to the complicated shape of electronic parts and the variation of film coefficients. Satisfactory preliminary designs can usually be achieved by "lumping" the problem and treating it as a whole. By constructing a simulated bread board model and conducting electrical and thermal tests, the system may be modified as required in order to obtain final design data. The tests on the bread board model should provide the temperature limits through which the liquid must be maintained for variations in the thermal environment of the heat exchanger. In order to operate within these temperature limits it may be necessary to add control equipment.

(2) Control of Cooling Systems

Direct forced liquid cooling systems should be designed for operation at the maximum cooling conditions. Under less severe thermal conditions it will be necessary to operate at slightly reduced capacity. The capacity of the system should be reduced as a function of the cooling demand. This may be accomplished by utilizing a temperature sensing element to actuate a control and vary the coolant temperature or flow rate in accord with the cooling needs. The degree of control depends upon the requirements of the most temperature sensitive part or parts in the electronic assembly.

(a) The simplest form of cooling rate control is the intermittent type. This function can be provided by a thermostat or temperature sensing device, located inside the electronic equipment, which turns the circulating pump on or off as required. Improved temperature regulation can be obtained by allowing the pump to run continuously and incorporating a thermostat to control a by-pass valve connected across the heat exchanger. Thus, the pump continues to circulate the liquid through the electronic equipment and the thermostat controls the flow of coolant through the heat exchanger to control the temperature of the coolant.

(b) For electronic equipments which require more precise temperature control than the intermittent control can provide, a step function type of control can be employed. The pressure and rate of flow of coolant may be varied in predetermined steps as desired. Several thermostats arranged to operate at various temperatures can be connected to short circuit portions of a tapped resistor in series with the pump motor. As the temperature of the electronic assembly decreases below the desired operating point the thermostats can introduce more resistance in series with the pump motor to reduce the coolant flow.

F. INDIRECT FORCED LIQUID COOLING SYSTEMS

1. General

In indirect liquid cooling systems the coolant liquid does not come in direct contact with the electronic parts. The primary cooling mode from the electronic parts to the coolant fluid is accomplished by other suitable means and the heat is transferred to the coolant.

Thus, the coolant receives the heat from the heat producing parts indirectly and carries the heat away to the sink. The electronic equipment may be internally designed to emphasize any one, or combinations, of the various methods of heat removal. Further, an indirect system may be applied to existing electronic equipment in order to improve operation.

2. Liquid Cooled Plates

Cold plate heat exchangers using fresh water as the coolant are recommended for shipboard application. Section VI discusses these exchangers in conjunction with forced air cooling. The use of fresh water can be much more effective than air because the film coefficient is greater (as much as 100 times). Another advantage in using water is that its specific heat is more than four times that of air so that for the same temperature rise the same weight of water will absorb four times as much heat. Further, water is rather easy to transport, high flow velocities may be obtained without excessive noise and, since it is over 800 times as heavy as air, the piping will be much smaller than equivalent air ducting. Thus, it appears that the water cooling of electronic equipment has considerable merit, provided that water supply and return piping is made available in each compartment.

3. Improvement of Existing Equipment by Liquid Cooling

Some shipboard equipments which were originally designed for forced air cooling are operating at temperatures well beyond safe values for reliability. In general, this situation has been caused by installation in confined spaces along with other equipments of high heat dissipation. Often the space is uninhabitable because the air ventilation system is inadequate for the heat load and no space is available for additional air cooling ducts. During the interim period, until more effectively cooled equipments are provided, it is recommended that the cooling of such equipments be supplemented through the utilization of fresh water cooled panel heat exchangers intimately attached to the inside of the equipment enclosures. When water is circulated through the cold panel, the original metal enclosure will act as an additional heat exchanger surface. The internal air should be cooled by the exchanger and recirculated over the equipment. Louvers and openings in the enclosure should be covered so that none of the internal air leaves the system. This will alter the free and forced convection air currents inside the equipment so that it may become necessary to divert the air flow along new paths to move the heat from the parts to the surface of the enclosure. Increasing the rate of circulation of the air by the addition of new fans or redirecting the output of existing fans should create more uniform internal temperatures.

4. New Equipment

It is recommended for new equipments that the individual components or subassemblies be designed for base cooling with a path of low

NOTE: SUBASSEMBLIES ARE SHOWN WIDELY SEPARATED FOR CLARITY

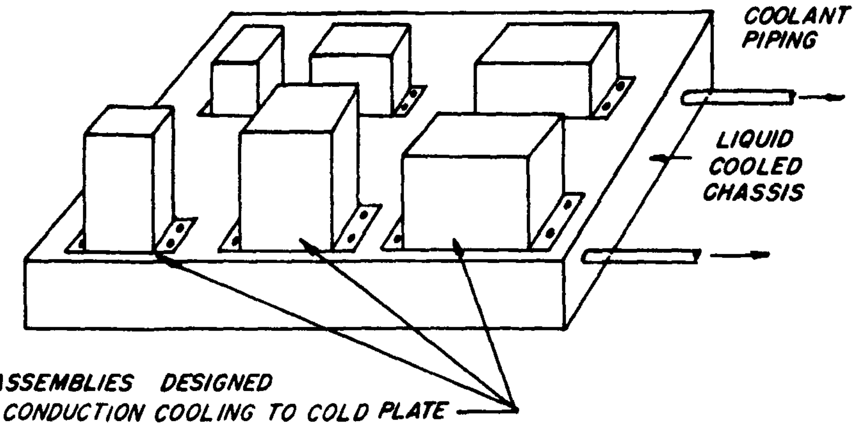


Fig. 44

LIQUID COOLED COLD PLATE

thermal resistance from the heat producing parts to the base of the subassembly. The base of the subassembly should be made so that it can be attached to a liquid cooled cold chassis (See Fig. 44). The optimum thermal shape would be a thin rectangular subassembly, like a shallow cigar box, which would slide in between two cold plates in sandwich fashion. The subassembly would thus obtain the greatest amount of cooling on the sides with the largest area.

The limitation of this method is the contact resistance between the cold plate and subassembly. Excellent cooling could be achieved through the use of liquid filled subassemblies completely immersed in the secondary liquid coolants.

Fresh water is recommended as the coolant for these systems. It is readily available on most shipboard and land installations. Another coolant liquid or refrigerant could be used. However, to avoid condensation of moisture on the electronic equipment, it is necessary that the coolant temperature be maintained ten or more degrees C higher than the dew point of the air surrounding the equipment. A sea water - fresh water heat exchanger tends to maintain the coolant above the sea water temperature.

With a shipboard fresh water cooling system the operating temperatures of the various heat producing parts in the electronic equipment can be maintained at almost any desired value above 50°C. Thus, air temperatures within an average equipment may be less than 125°C, while the coolant from a sea water - fresh water heat exchanger would seldom be lower than 35°C.

During the winter when the sea water is near 0°C it may be necessary to control the flow of fresh water through the sea water to fresh water heat exchanger with a bypass valve in order to prevent over cooling or possible freezing. Such a control will also aid in reducing equipment warmup time.

On land installations water from a well would probably be at about 15°C. This could cause condensation in the electronic equipment unless a proper balance between air and coolant temperature is maintained. If the land based system utilizes a closed liquid system and rejects heat to the atmosphere through a liquid-to-air heat exchanger, the same type of control of dew point could be maintained as in the sea water to fresh water heat exchanger.

Note: All water cooled equipment should be provided with drains to alleviate freezing during non-operating periods. Further, liquid level devices should be included in order to insure that the equipment is adequately filled with water and the piping should be arranged so that "air locks" are minimized.

G. COMPOSITE LIQUID COOLING SYSTEMS

Composite systems, by definition, are those which incorporate both direct and indirect liquid cooling. Each part of such a system must be treated according to its type. In a typical composite liquid cooling system, the heat producing electronic parts are directly immersed in a suitable liquid (primary coolant). A low pressure pump circulates the liquid through the system. The heat is transferred at a heat exchanger to another liquid (secondary coolant). This liquid is then circulated to another heat exchanger to deliver the heat to the sink.

H. CHARACTERISTICS OF COOLANTS

1. Silicone Fluids

a. Straight dimethyl fluids are known as the 200 series silicone fluids. It is not recommended that these fluids be operated at temperatures exceeding 150°C for optimum heat stability in direct liquid cooled equipments. The 200 series fluids differ from each other only in viscosity. Viscosities ranging from .65 to 10^6 centistokes* are available by blending.

b. The 500 and 700 series silicone fluids are blends of the dimethyl and phenolmethyl fluids. They are stable to 200°C for direct liquid cooling applications. These fluids are not available in as wide a range of viscosities as the 200 series fluids.

2. Other Coolants

The characteristics of other coolants are given in Appendix C. Only those fluids which are recommended for electronic heat removal applications are noted therein.

* The "centistoke" is the unit of kinematic viscosity in the c.g.s. system of units, and the kinematic viscosity is equal to the absolute viscosity divided by the mass density. However, in engineering heat transfer as covered in this Manual, the units of absolute viscosity are lbs./(ft.)(hr.). To convert to lbs./(ft.)(hr.), multiply centistokes by 2.42 times the mass density in gms./cc. Note that in the c.g.s. system, mass density and specific gravity relative to water are numerically equal.

VIII. VAPORIZATION COOLING

A. GENERAL

In vaporization cooling heat is removed from the electronic parts by a change in state of the coolant from a liquid to a vapor. For a given weight of coolant, vaporization cooling provides the most effective cooling of any method.

When a unit mass of liquid is heated at constant pressure it absorbs heat equal to its specific heat (at constant pressure) for each degree rise in temperature. Water, for example, has a specific heat of 1.0 Btu/(lb.)(°F) and when heated, from say 100° to 150°F, absorbs 1.0x(150-100) or 50 Btu per pound. This heat absorbed by the liquid to raise its temperature is termed "sensible heat". If the liquid continues to absorb heat, it will ultimately reach a temperature at which boiling or vaporizing will begin. The term "saturated vapor" is applied to a liquid which has reached the boiling temperature corresponding to its vapor pressure. The saturation temperature is dependent on the pressure and increases with pressure. For example, water at atmospheric pressure (14.7 psia) boils at 212°F. If the pressure is increased to 40.0 psia it boils at 267.2°F and if decreased to 5 psia it boils at 162.2°F.

When the heating of the saturated liquid is continued at constant pressure, the ensuing boiling will take place at constant temperature and will continue until no liquid remains. The vapor (steam in the case of water) is in equilibrium with the liquid at a constant pressure and temperature during this vaporizing or boiling process. When the liquid is completely vaporized it is called "saturated vapor", the phase change having gone to completion.

The amount of heat required to vaporize completely a unit weight of saturated liquid is known as "the heat of vaporization" or "the latent heat of vaporization". Its value depends on the saturation pressure or temperature and decreases as the pressure increases. Compared with the sensible heat, the heat of vaporization is much larger. For example, 970.3 Btu (284 watt hours) are required to completely vaporize one pound of water at a saturated pressure of 14.7 psia and 212°F. Water has the highest heat of vaporization of most liquids and, this, together with its very high heat transfer coefficient of vaporization, makes it a superior heat transfer medium.

Similar to liquid cooling, vaporization cooling may be classified into direct and indirect systems. In a direct system the electronic parts may be completely immersed in a volatile refrigerant. If the surface temperature of a heat producing part exceeds the boiling point for a given pressure, small bubbles will be formed at the surface, thus removing heat equal to the latent heat of vaporization of the weight of liquid vaporized.

A direct vaporization cooling system can be very effective since the heat is removed directly from the surface of the part. In the direct systems, since the electronic parts are exposed to the coolant, they are subject to any long term corrosive or solubility effects which may exist. The coolant must be chemically inert with respect to, and compatible with, the electronic parts.

In a direct cooling system the vaporized refrigerant liquid is considered to be the primary coolant. In an indirect system the initial heat is removed from the electronic parts to a separate heat exchanger by any suitable means. The coolant is only used to accomplish cooling in the heat exchanger (secondary coolant).

B. NOTES ON VAPORIZATION COOLING AND BOILING

If saturated vapor is further heated at constant pressure, its temperature rises and the vapor becomes "superheated". The specific heat of the vapor is smaller than that of the liquid. Superheated steam has a constant pressure specific heat of about 0.49 Btu/(lb.)(^{o}F) at atmospheric pressure. In vaporization cooling of electronic equipment, it is doubtful that superheated vapor will be encountered, since the saturated vapor must again come in contact with a hot surface away from the liquid and the small specific heat and low coefficient of heat transfer do not justify using superheated vapor as a heat transfer medium.

The many variables involved in the mode of heat transfer known as boiling has complicated the formulation of general equations from which boiling coefficients can be predicted. The following is from reference(22):

> "When a liquid is boiled, as the temperature difference between the boiling liquid and surface is increased, the rate of boiling (and the amount of heat flowing) increases, but reaches a maximum with a critical temperature difference above which the rate of boiling decreases. As the temperature difference becomes greater than this critical difference, the vapor formed by boiling acts as an insulator, impeding the transfer of heat. The critical difference for water is about 45^{o}F.
>
> "In general, the rate of boiling is increased as the surface becomes rougher, and as the liquid being boiled is agitated. It is increased with an increase in temperature difference up to the critical point, and is reduced by scale and dirt deposits. For boiling, it is usually necessary to measure the overall transfer coefficients for the particular liquid and physical conditions in question, as there are few data available."

C. DIRECT VAPORIZATION COOLING SYSTEMS

1. Liquid Potting

An example of liquid potting, which is one of the simple vaporization cooling systems, is shown in Figure 45. The electronic parts are completely immersed in a volatile refrigerant, such as Freon "113". Since the container is hermetically sealed, a small space has been left in the top of the container to allow for expansion of the Freon vapor. When the surface above this space is cooled, the vapor will condense and return to the liquid state. The heat producing components are mounted with their longest axis in the vertical position to aid the formation of convective currents in the liquid. Heat is rejected not only at the top surface of the container by condensation, but also at the sides and bottom by convection. As the surface of the heat producing parts exceeds the boiling temperature of the Freon (which depends upon the internal pressure in the hermetically sealed container), bubbles are formed at the hot surfaces, submerged boiling takes place and induces free convective currents in the Freon. Due to the high film coefficient of the boiling liquid, the temperature difference between the parts and the liquid is usually very small.

a. Design Considerations

One of the primary design considerations in liquid potting is the necessity to design the container so that it can withstand high internal pressures with an adequate factor of safety. For example, Freon "113", at a saturation temperature of 122°C, has an absolute pressure of 70 pounds per square inch. Care should be taken in subjecting electronic parts to high pressures. Reference to a Mollier diagram will provide the operating pressure for a given saturation temperature.

The temperature of the liquid will vary with the internal vapor pressure until equilibrium with the thermal environment external to the container is reached. Experiments show that temperature differences between components and containers may be as low as from 5 to 10°C. The most temperature sensitive component part will limit the maximum fluid temperature to a value that will still maintain circuit stability. For this reason, the thermal advantages of the small fluid-to-part temperature differences produced by direct vaporization cooling are desirable.

The practical considerations of the internal pressure on electronic parts and the heat dissipation capabilities of the container to its thermal environment define the conditions under which this type of cooling may be utilized.

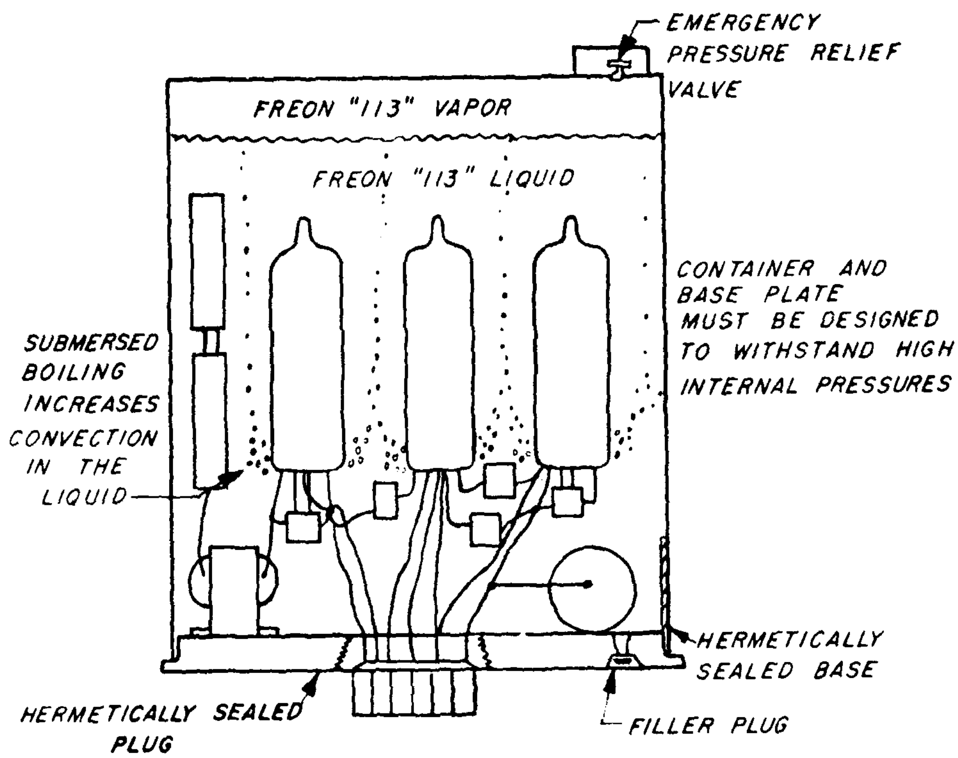


Fig. 45

VAPORIZATION COOLED
LIQUID POTTED SUBASSEMBLY

2. Expendable Direct Vaporization Cooling

In expendable vaporization cooling systems the heat bearing vapor is expended to the atmosphere as a waste product. Figure 46 presents an example of such a system in which the electronic parts are completely immersed in a suitable refrigerant liquid. In order to expend the heat laden vapor to the atmosphere, it is necessary to operate the subassembly at a temperature that will produce a refrigerant vapor pressure greater than atmospheric pressure.

a. Control of Temperature and Pressure

There are two ways to control the temperature in expendable systems:

(1) A constant pressure may be maintained above the liquid by the use of a spring loaded pressure relief valve. A special valve has been developed for expendable systems to maintain an evaporating liquid at a constant absolute pressure above an evaporating liquid (See Ref.23).

(2) The second method is to use a temperature sensing device to control a variable area discharge nozzle which varies the internal pressure and maintains the temperature sensing device at a constant temperature.

The cooling capacity and operating time of an expendable system is limited by the supply of coolant available. It is necessary that the heat laden and, perhaps, toxic vapor be directed to the outside atmosphere and not be released in any space occupied by personnel.

b. Example 7 - Computation for an Expendable Vaporization Cooling System

An approximation of the amount of refrigerant required to dissipate a given power may be computed as shown in the following example:

Given: An electronic subassembly at 32°C ambient, dissipating 10 watts in a 2 1/2" x 2 1/2" x 4" container, half of which is occupied by electronic parts; the remaining volume is filled with Freon "113".

Assume: All of the heat dissipated vaporizes the Freon "113", which is expended through the relief valve to the atmosphere. (There is no heat transfer from the case).

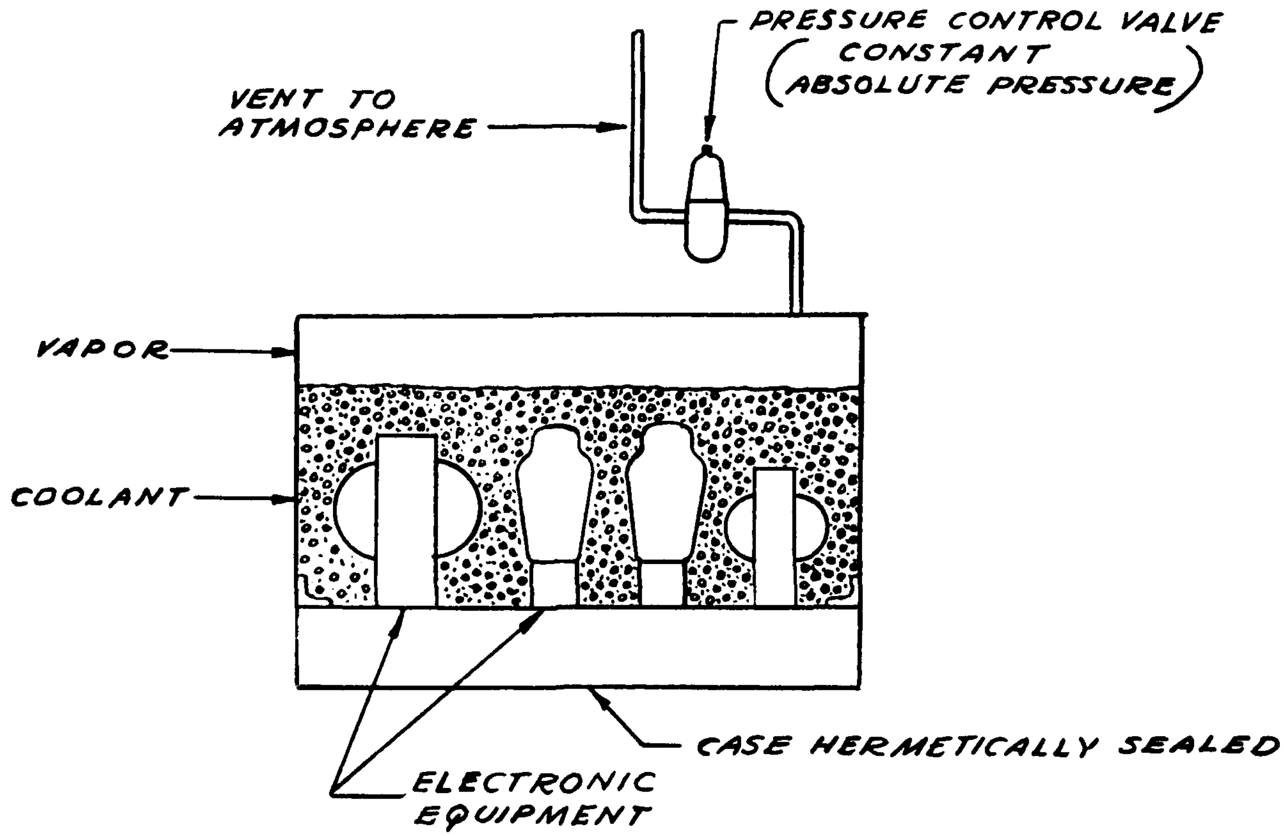


Fig. 46 TYPICAL DIRECT EXPENDABLE VAPORIZATION COOLED ELECTRONIC DEVICE

The system operates as a direct expendable vaporization cooling system in which the pressure relief valve operating pressure is 18 psia, corresponding to a 54.2°C saturation temperature.

Required: The quantity of Freon "113" required and the time the subassembly will operate without requiring replenishment of the Freon.

The "sensible heat" absorbed by the liquid Freon "113" at 32°C (90°F) heated to saturated liquid at 54.2°C (128°F) is:

Freon Tables (Appendix C). The specific heat of Freon "113" at an average temperature of 109°F = 0.223

Sensible heat = 0.223 x (128-90) = 8.5 Btu/lb.

The heat absorbed by the Freon "113" due to the change from the liquid to vapor state at 54.2°C (128°F) is:

Latent heat of vaporization = Enthalpy of saturated vapor minus Enthalpy of saturated liquid at 18 psia.

Latent heat = 95.0 - 35.8 = 59.2 Btu/lb.

Total heat absorbed = sensible heat + latent heat of vaporization = 8.5 + 59.2 = 67.7 Btu/lb.

Power dissipation in Btu/hr.

1 watt = 3.413 Btu/hr.

10 watts x 3.413 = 34.13 Btu/hr.

Pounds of Freon "113" required per hour of operation lbs./hr. required = $\frac{34.13}{67.7}$ = .504 lbs. required to cool subassembly.

Number of hours the subassembly, full of Freon will last:

Total cu. content = $\frac{2.5 \times 2.5 \times 4}{1728}$ = 0.0145 cu.ft.

Liquid volume $\frac{0.0145}{2}$ = 0.00725 cu. ft.

Density of liquid Freon "113" approx. 97 lbs./cu. ft.

Pounds of Freon in subassembly = 0.00725 x 97 - 0.704 lbs,

Duration of Freon in subassembly $\frac{0.704}{0.504}$ = 1.4 hrs.

3. Direct Spray Systems

a. General

Spray cooling can be used to provide improved film coefficients over those obtained with direct immersion. Further, the total weight of coolant can be greatly reduced. Fig. 47 shows a direct evaporative spray cooling system. The refrigerant is pumped from the reservoir into the spray manifold, from which it is sprayed over the heat producing parts in the subassembly. The surplus refrigerant is collected in the bottom of the subassembly and returned to the reservoir for recirculation. This direct spray system is more economical of the weight and quantity of fluid used for a given amount of cooling than other direct systems, and may be operated at atmospheric pressure. A reasonably high pump pressure is required to overcome the pressure drop in the system.

b. Design Considerations

One of the main problems in a spray system is in obtaining complete wetting of the heat producing electronic parts. This is dependent upon the pump pressure, position and pattern of the spray jets. More uniform wetting of tubes can be obtained by wrapping copper screens and wick material around the parts to be cooled. The pump power requirements in this system may be rather high since a large positive pressure must be maintained in the manifold. All of the other factors which apply to the spray cooling system described in the liquid cooling section also apply to vaporization spray systems.

D. INDIRECT VAPORIZATION COOLING

1. General

In indirect vaporization cooling systems, the electronic subassemblies may be constructed and assembled to employ any adequate primary cooling means. The waste heat is removed from the surface of the subassemblies and transferred to a heat exchanger for rejection to an evaporative coolant. Such an arrangement permits the use of a coolant which is not compatible with the electronic parts.

A large cooling capacity may be obtained with a very small expenditure of power if some readily available liquid, such as water, is used as the evaporative coolant. This means that the water, at atmospheric pressure, would have to be heated to 100°C in the heat exchanger and the steam rejected to the

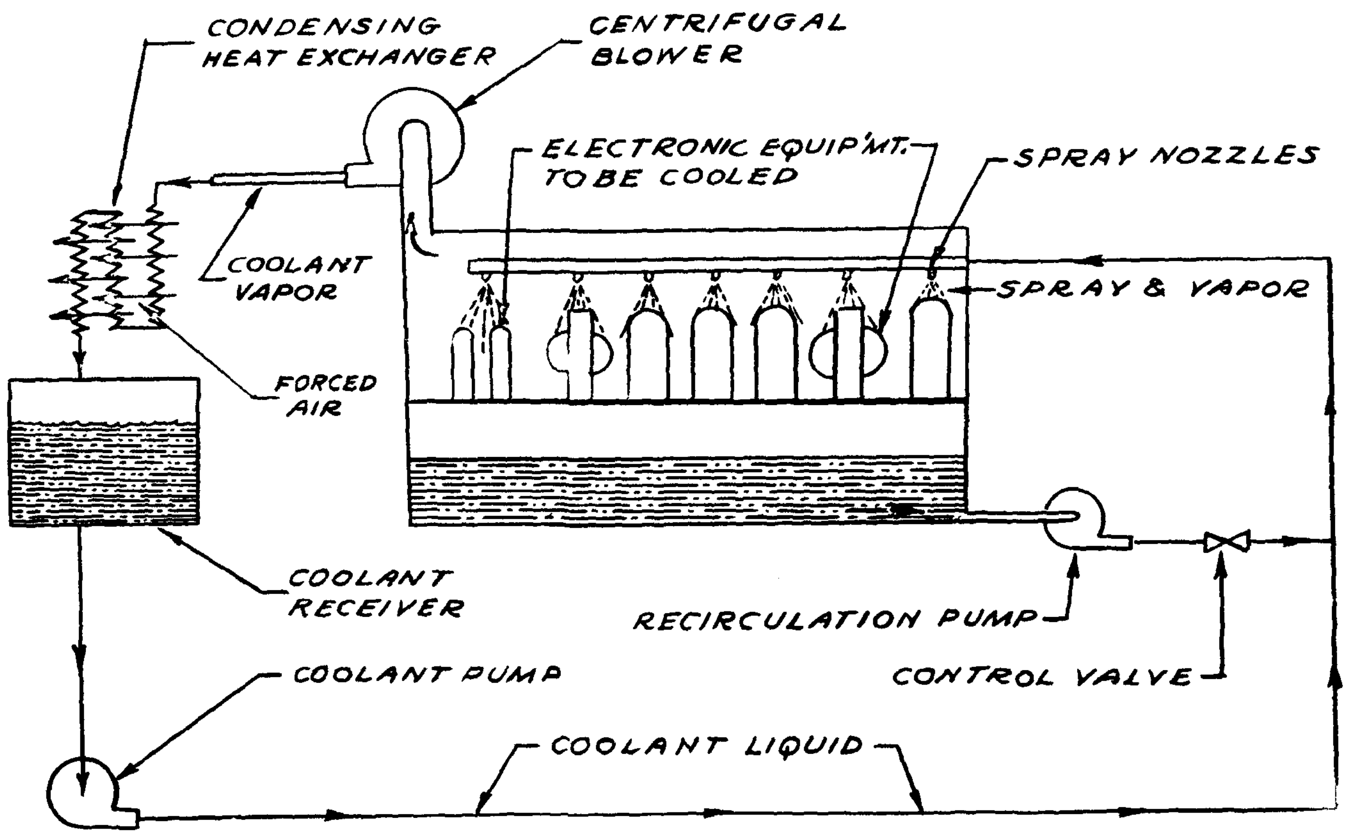


Fig. 47 DIRECT EVAPORATIVE SPRAY COOLING SYSTEM

atmosphere. The steam expended could be replaced by the addition of more water to the system, thereby eliminating the equipment required to condense the vapor.

If liquid cooling is used as a primary cooling means, the rate of primary coolant flow through the exchanger may be varied by throttling or by using a by-pass valve across the heat exchanger. If an air-to-liquid heat exchanger is used, the blower may be throttled to control the velocity through the exchanger and the supply temperature may be controlled by a by-pass around the exchanger.

2. Examples of Indirect Vaporization Cooling Systems

a. Liquid-to-Liquid Systems

Figure 48 shows a typical expendable indirect system in which the liquid potted electronic subassemblies are directly immersed in the secondary refrigerant liquid. The subassembly cases act as the heat exchanger between the primary potting liquid and the secondary evaporative refrigerant. The evaporating temperature of the coolant is controlled by the pressure relief valve through which the refrigerant vapor is vented to the atmosphere. When the liquid level falls below a certain level the pump control switch actuates the electric pump to replace the vaporized refrigerant. The check valve allows the pump to replace the vaporized refrigerant as required but does not permit the refrigerant to leak back when the pump is not in operation. If the liquid is stored at a higher pressure than in the heat exchanger, a pump will not be required and the switch could actuate a demand valve on the reservoir to supply additional refrigerant as needed.

Figure 49 presents two equipments that are cooled by direct forced liquid cooling. The primary coolant is pumped through the subassemblies by means of the primary coolant pump, and the flow control valves in the discharge lines of the subassemblies control the rate of cooling. The heat bearing primary coolant fluid is in turn cooled by the evaporating liquid in the heat exchanger. The saturation temperature in the heat exchanger is regulated by the constant pressure relief valve. In this system, the primary coolant is a liquid which is capable of maintaining its liquid state at a much higher temperature than the secondary evaporative coolant. This system, as shown, is of limited operating time, as vaporization of the secondary coolant continues only until all of the liquid is spent. It may be made continuous (closed cycle) by application of a condenser and reservoir.

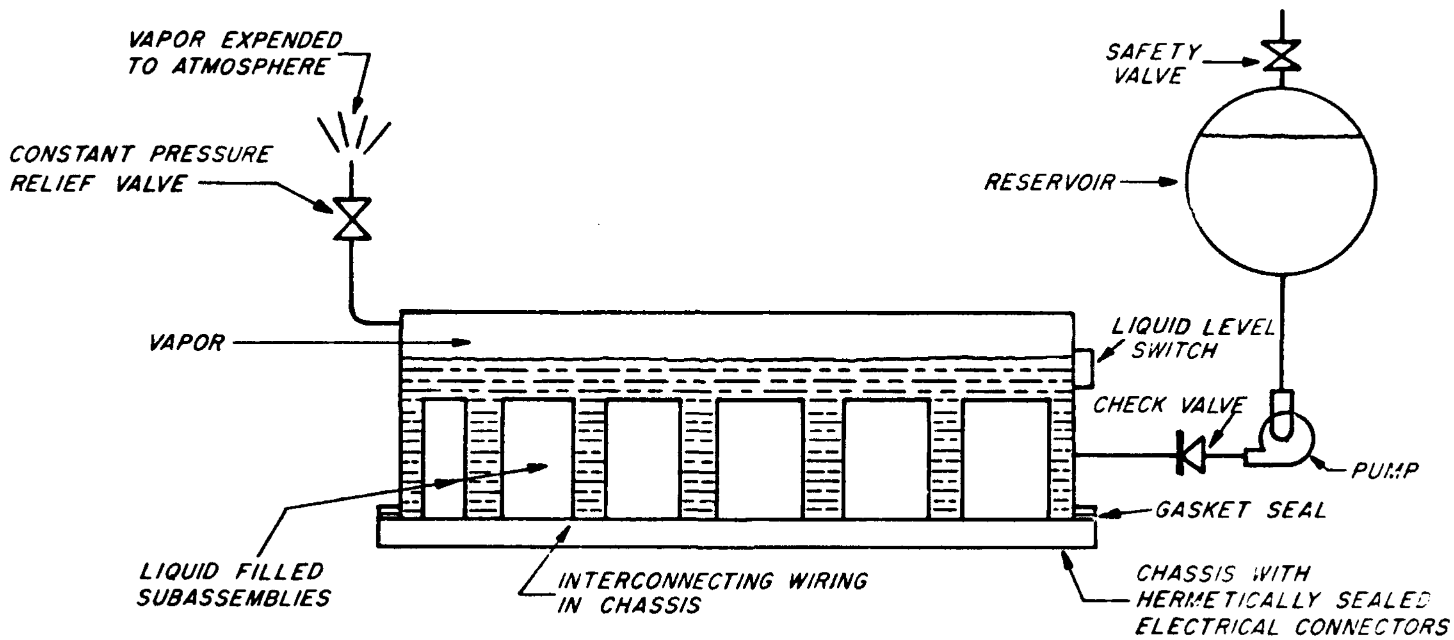


Fig. 48 LIQUID TO LIQUID VAPORIZATION COOLING SYSTEM

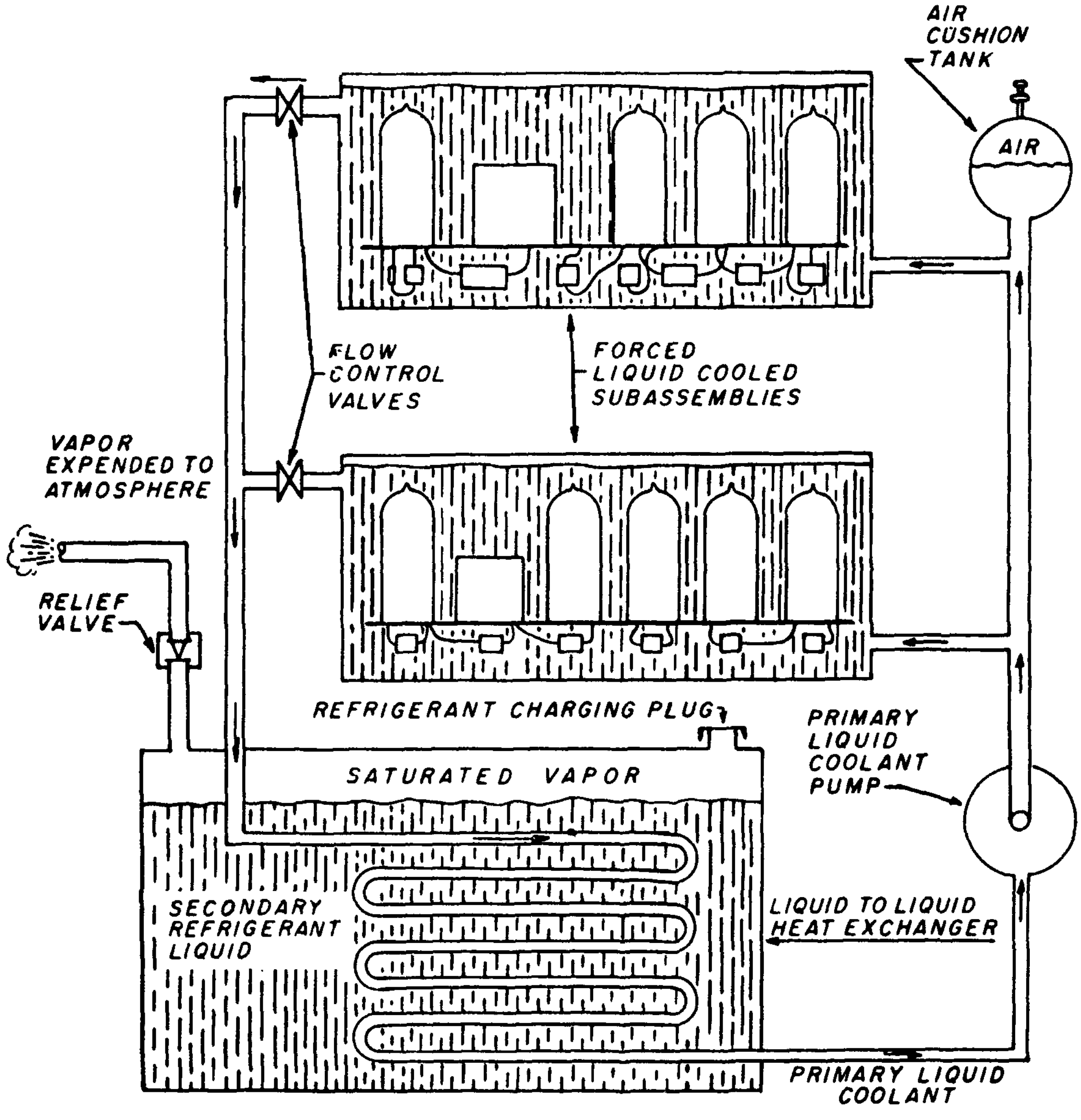


Fig. 49 LIQUID TO LIQUID INDIRECT VAPORIZATION COOLING SYSTEM

b. Air-To-Liquid Systems

Indirect expendable vaporization cooling systems may be employed in conjunction with forced air cooled equipment. The cooling air, in such a system, is passed through a heat exchanger cooled by the evaporation of a suitable refrigerant. The air temperature at the entrance to the electronic equipment will, of course, depend upon the temperature of the refrigerant used in the heat exchanger.

E. Design Notes

Design data applicable to cooling of electronic equipment by vaporizing liquids are meager. This makes initial analysis of a given design only practical in broad and general terms. Thermodynamic properties of refrigerants may be obtained from Mollier diagrams. See Appendix C.

Only after thermal and electrical evaluation tests of a prototype model can the most effective conditions of operation be determined. Generally speaking, heat transfer by vaporizing a liquid is influenced by the following factors:

1. A high film coefficient and good wetting properties are desired in the coolant, since the wettability of the components' surface influences the size and shape of the vapor bubbles which are formed and liberated directly from the surface of the heat producing parts.

2. Small bubbles which form on surfaces that are easily wetted are more promptly disengaged from the surface than larger bubbles.

3. Ease of bubble formation and ease of detachment of the bubbles from the hottest surface promotes high film conductance.

4. The liquid in direct contact with the heat producing part is usually superheated several degrees above the saturation temperature that corresponds to that liquid.

5. Rough surfaces of electronic parts show less superheating of the liquid and better heat transmission than smooth surfaces.

Until extensive empirical design information is made available, it will be necessary to use the experimental approach in the design of each new vaporization cooling system. The selection of the pump presents a problem. In some applications the pump should be capable of pumping either liquid or vapor without the need of a lubricant in the refrigerant. The amount of refrigerant in the system and its distribution over the electronic parts must be determined experimentally during initial electrical and thermal testing of the equipment.

The technique of sealing refrigerants is new to the electronics field but not to the refrigeration field. One difficulty encountered has been in attempting to make a pressure tight seal with a flat rubber gasket between two flat surfaces. The recommended method for a pressure seal is to form a groove in the two flat surfaces to receive a recessed gasket or O-ring. With this type of seal, the plates do not have to be brought together with as much uniformity, and the equipment can be opened numerous times for inspection without damage to the gasket or the effectiveness of the seal.

F. NOTES ON PROPERTIES OF VAPORIZATION COOLANTS

The physical and electrical characteristics of various vaporization coolants are presented in Appendix C. Supplemental data may be obtained from the manufacturers.

1. Freons

Freon "113" has been found to be the most practical coolant for direct vaporization cooling systems because of its compatibility with most materials used in electronic equipment. However, it has relatively high vapor pressure and relatively low latent heat. The heat transfer coefficient for Freon-12 boiling within tubes is from 150 to 300 Btu/(hr.)(sq.ft.)(^{o}F) difference between fluid and tube wall temperatures. It appears that these figures are representative of the family of Freon refrigerants although little data is available. These values are low when compared to boiling water coefficients which may exceed 2000.

2. Perfluorocarbon Liquids

Where the "Freon" refrigerants are unsuitable as vaporization coolant liquids because of too low a boiling point at atmospheric pressure, the lower chemical series perfluorocarbons may find successful application.

"Completely fluorinated hydrocarbons, that is, compounds consisting of carbon and fluorine, have been given the generic name, "perfluorocarbons". The perfluorocarbons constitute a class of stable and chemically inert compounds. The lower members are thermally stable up to temperatures of 400^{o}C. They will not burn and are resistant to the action of concentrated acids, at temperatures well above 100^{o}C. They do, however, decompose at a dull red heat. They are attacked only by such reagents as metallic sodium and potassium at temperatures of 200^{o}C and at lower temperatures by elemental fluorine. The higher chemical series members exhibit similar chemical inertness but are somewhat more heat sensitive."

"The perfluorocarbons are characterized by low indices of refraction, high specific gravities and lower boiling points than other compounds of similar molecular weight. They are insoluble in polar solvents. They are slightly soluble in non-polar solvents, but much less so than are hydrocarbons, esters, chlorinated compounds, etc.

"The higher members of the perfluorocarbon liquid series tend to be substantially insoluble, less than 0.1%, in all organic solvents. An exception to this is Freon "113" ($C_2F_3Cl_3$) in which these perfluorocarbons are miscible. It is evident from the table of solubilities that insolubility increases with molecular weight (ref. 24)."

Electrical Properties

"The perfluorocarbons are characterized by low dielectric constants, which, in general, change only slightly over the frequency range of 100 cps. to 100 kc. Thus, the dielectric constant of C_7F_{14}, the member with the lowest value, ranges from 1.69 to 1.70 over the above frequency range, while a perfluorocarbon with a boiling range of 210 to 240°C has the highest value and the greatest range, 2.02 at 100 cps. to 1.90 at 100 kc.

"The power factor is, in general, low and compares favorably with standard transformer oils. Apparently no relationship exists between power factor and structure of the molecule. Thus, C_7F_{14}(perfluoromethylcyclohexane), the simplest member of the series has a power factor varying from 0.0045 at 100 cps to 0.0005 at 100 kc. while the value for perfluorocarbons distilling from 130 to 150°C (a complex mixture) varies from 0.0015 to 0.0003 over the same frequency range...

"The volume resistivity and the dielectric strength of the perfluorocarbons are generally much better than standard transformer oils. Thus, C_8F_{16} has a volume resistivity of 1.2×10^{12} ohms and a dielectric strength of 15,000 volts as compared to the values for a standard transformer oil of 1.1×10^{12} and 15,000. The opposite extreme is that of perfluorocarbons distilling from 210 to 240°C, which have a resistivity of 1.7×10^{14} ohms and a dielectric strength of 20,000 volts. Here again, little relationship exists between structure and volume resistivity and dielectric strength." (Ref.24).

3. Fluorochemicals

Fluorochemical liquids which possess chemical and thermal stability are now being offered commercially. They are nonflammable, nonexplosive, nontoxic, noncorrosive to metals, plastics, etc., and have excellent insulation properties. These materials possess a combination of properties which make them of particular interest to the heat transfer designer. They offer the best present hope for coolants applicable to operation from subzero to 400°C or higher. When properly applied, it would appear that they can very markedly increase the efficiency of certain heat transfer operations, (ref. 25).

"Fluorochemical liquids are remarkably stable in that they are not attacked by concentrated acids, solid alkalies, strong oxidizing agents or reducing agents. Two fluorochemicals, perfluorotributylamine, $(C_4F_9)_3N$, Fluorochemical N-43, and a perfluoro cyclic ether, with an empirical formula $C_8F_{16}O$, Fluorochemical 0-75, are representative of the classes which are available. The thermal stability of Fluorochemical 0-75 was tested by heating at 400°C (725°F) for 65 hours in a stainless steel autoclave in the presence of copper. There was no detectable decomposition. Fluorochemical N-43 under the same conditions decomposed less than 0.4% as determined by free fluoride ion. The liquid became faintly straw-colored and there was a thin dark deposit on the copper. This is similar to other tests which showed the perfluoro tertiary amines to be slightly less stable than the perfluoro ethers. In another series of tests to determine effects on common materials of construction, test strips of iron, copper, aluminum, silicone rubber, varnished cambric, Teflon, and Kraft paper were partially submerged in Fluorochemical N-43 and Fluorochemical 0-75 at 90°C (194°F) for eleven weeks. No change was noted on the test strips or the liquids. In similar tests, where excess water but no air was present, there was only slight corrosion of the metals. The corrosion was barely discernible in the liquid-phase portion of the strip, which can probably be accounted for by the low solubility of water in fluorinated liquids (less than 25 ppm). In the case of varnished cambric, the water extracted the varnish. Because of the insolubility of fluorinated liquids in organic solvents, they do not swell rubber nor extract the plasticizers from insulating materials or sealants. Teflon is not affected by either of the above liquids at temperatures up to 250°C." (Ref. 25)

The electrical properties of the two fluorochemical liquids are of particular interest. Both liquids have low dielectric constants and low loss factors over wide frequency and temperature ranges. See Figs. 57, 58, 59 and 60 in Appendix C.

Both the perfluoro liquids have high dielectric strength and high resistivities as shown in Table 11.

TABLE 11

Resistivity and Dielectric Strength of Fluorochemicals

Designation	N-43	0-75
Resistivity (2 kv/cm) ohm-cm	$10^{14} - 10^{16}$	$10^{15} - 10^{17}$
Dielectric strength, ASTM D-877	45	43

The dielectric strength of the vapors is also high. At one atmosphere pressure, the value approaches that of the liquid.

Two methods of heat transfer in which the use of fluorochemical liquids is advantageous are by free convection and evaporative spray cooling. In the former, the high density, high coefficient of expansion and low viscosity of these liquids result in high free convection modulus, while the low specific heat, is compensated by low thermal conductivity.

The modulus for fluorochemical N-43 ranges from 5 to 2000 x 10^8 per cu. ft. - °F over the temperature range 77° to 350°F. The modulus for fluorochemical 0-75 ranges from 40 to 200 x 10^8 per cu. ft. - °F from 77° to 214°F, while a transformer oil ranges from 1 to 20 x 10^8 per cu. ft. - °F between the latter temperatures. See Tables 29 and 30.

4. OTHER COOLANTS

Table 31 from Reference 26 lists properties of eleven coolants according to their latent heat. The first six refrigerants merit consideration for direct vaporization cooling because of their high dielectric constants. With the exception of Freon "113" most of them have some deleterious effect on certain materials used in electronic equipment.

The last five refrigerants in Table 31 are listed for possible use in indirect vaporization cooling systems. The last two refrigerants listed are binary solutions of water and methanol and water and ammonia. Ref. 26 shows curves of the variation of absolute pressure for evaporation at constant temperature and the variation of temperature for evaporation at sea level pressure as functions of the latent heat added, for the two binary solutions. Curves are also shown that indicate the water content by weight of the remaining liquid coolant at any point of the evaporative process.

Temperature control of a boiling binary solution cannot be maintained by a constant absolute pressure above the liquid, as with a single substance, because the vapor pressure of the binary solution gradually decreases as the most volatile part of the solution boils away. The change in vapor pressure is also accompanied by a change in temperature.

IX. THE SELECTION OF OPTIMUM COOLING METHODS

A. GENERAL

This section is related to the relative merits of the various cooling means and methods of determining the most practical cooling mode for a particular application. The figure of merit which has been assigned is the heat concentration in watts/cu.in. In those instances wherein the external surface area of the device limits the thermal resistance, the unit heat dissipation in watts/sq. in. is also mentioned as a secondary figure of merit. These numbers can also be used as a measure of the magnitude of the design problem.

In general, there are a number of practical cooling techniques which are satisfactory within stated limits. Certain of these techniques with slight modification and some experimentation can be applied to specific designs. It is true that the thermal design of electronic equipment has not been reduced to an exact mathematical science. However, the order of magnitude of any specific cooling design can be determined. Since this places the design in a finite known range, the limits of the design and experimentation can readily be established.

The selection of the optimum cooling method should be preceded by the breadboard development of the electronic circuit. If the electronic performance is influenced by the cooling method, the circuit of the prototype model should be modified after the initial breadboard tests.

The selection of cooling methods discussed herein are primarily predicated upon the heat concentrations and signal frequencies involved in shipboard and ground based equipments. Other factors such as the complexity of the equipment, space, power, thermal environment, available sinks and cost must also be considered by the designer. Since the optimum method of heat removal within a subassembly and a unit may differ from that used to transfer the heat to the ultimate sink, each will be separately discussed.

B. HEAT TRANSFER WITHIN A UNIT

1. General

The method of heat removal from within a subassembly must be such as to provide a low temperature gradient between the heat producing parts and the cooled surface or the local sink. The cooling method must be simple, light weight, reliable, easily maintained and economical. Further, it should occupy a minimum of volume, preferably utilizing the voids between densely packaged parts.

2. Natural Methods

Natural cooling means are recommended for use within most miniaturized electronic subassemblies. They are frequently the only possible means of heat removal. Hermetic sealing and the dense packaging of parts can prevent the use of other techniques. Metallic conduction should be considered initially as the primary cooling means. Radiation cooling is not recommended as a primary means, since high temperature differences are required for appreciable heat transfer. Further, the control of the cooling path is lost since the heat will be radiated into nearby subassemblies. Convection cooling requires large areas which are seldom available within subassemblies. Also, convection currents will transfer the heat into other locations which will only require additional cooling.

Plastic embedment may be used at heat concentrations of the order of 0.25 watts per cubic inch at ambient temperatures of the order of 85°C. Metallic conduction can be used for heat concentrations as great as 2 watts per cubic in. (See Fig. 50) The maximum unit heat dissipation for free air cooled surfaces is usually 0.25 watts/sq. in. In a few high temperature devices, unit heat dissipations as high as 0.5 watts/sq. in. have been achieved. It should be noted that hermetic sealing is essential for many equipments which must operate under rigorous climatic and environmental conditions. Overall package sealing simplifies the problem to the extent that only one large seal must be made and all parts are protected by it. Admittedly, overall sealing prevents easy access for servicing by requiring the entire subassembly to be opened with consequent loss of any inert gas or liquid and the possible entrance of moisture. Sealed subassemblies can be easily replaced, especially if they are of the plug-in type. Certain electronic circuits such as RF, IF, and video amplifiers cannot readily tolerate the increased capacitance and losses associated with plastic embedment. In such instances an inert gas is advisable. Gases with high thermal conductivities such as helium or hydrogen can be used to increase heat transfer by gaseous conduction. Additional information on these matters is incorporated in Natural Cooling Methods Section V.

3. Forced Air

Forced air cooling is an excellent cooling method, which can be used if the spacing between parts within the subassembly is adequate for air flow. Considerable heat can be removed by this method. (See Fig. 50) Individual parts with heat dissipations as great as 2 watts per sq. in. can be cooled at high Reynolds numbers. However, the power required to force air over objects and through ducts and heat exchangers may be considerable. Further, the interchangability of forced air cooled subassemblies will be rather limited to a few special equipments which are provided with adequate fans and ducting for each subassembly. Further details are incorporated in Forced Air Cooling Section VI.

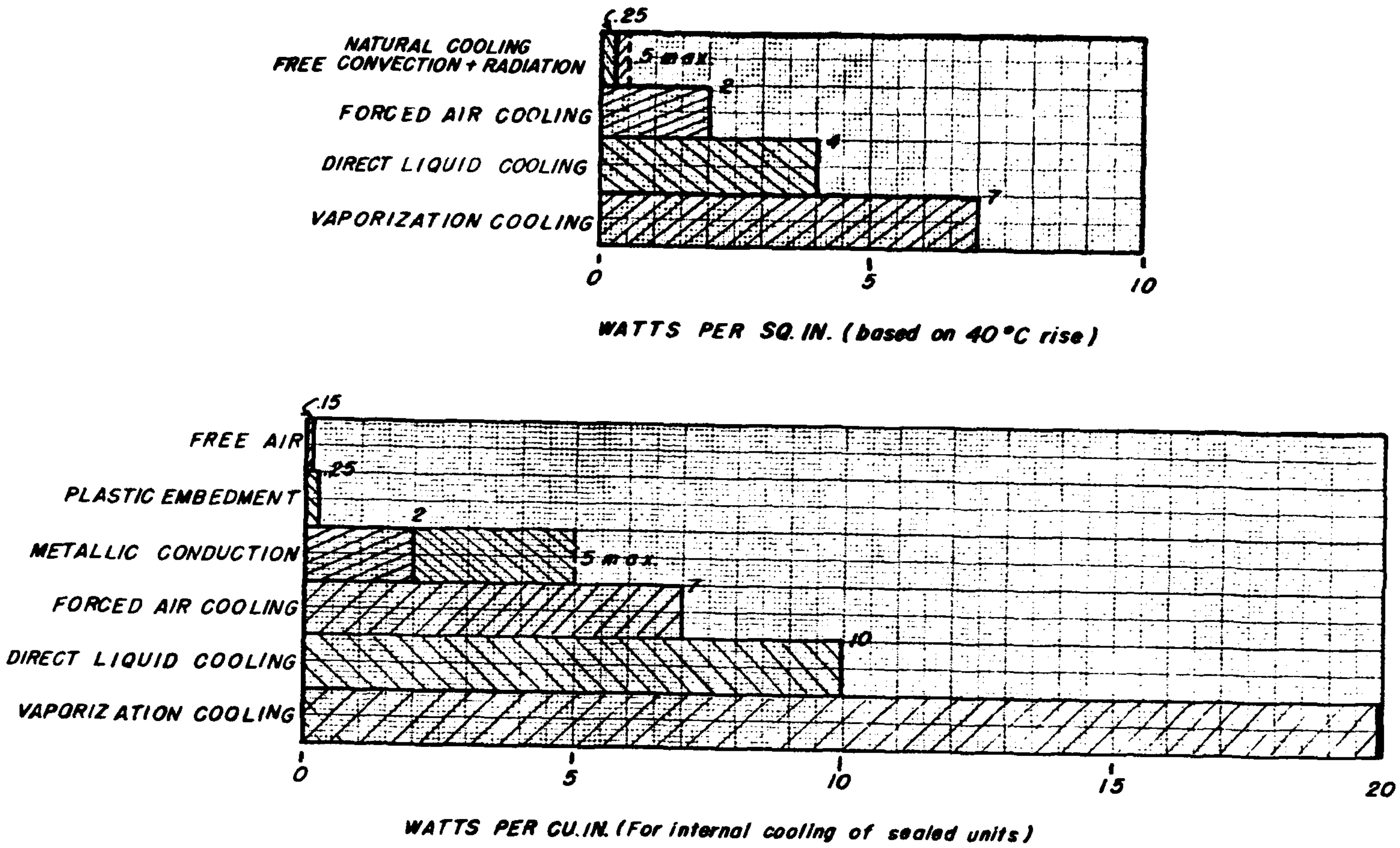


Fig. 50 - COMPARISON OF METHODS OF COOLING

4. Direct Liquid Cooling

This cooling means is particularly applicable to subassemblies having high heat concentrations or those which must operate in high temperature environments with small temperature gradients between parts and cooled surfaces. Unfortunately, direct liquid cooling can be used only in circuits which can tolerate the increased stray capacitance and electrical losses due to the high dielectric constant and power factor of liquids.

New equipments can be designed for several types of liquid cooling systems any one of which may have cooling capacities greater than that of forced air systems. (See Fig. 50) The cases of sealed sub-assemblies can be designed for direct immersion in the coolant (indirect liquid cooling) or the electronic sub-assembly can be filled with a liquid such as a silicone fluid (direct liquid cooling). Cooling of directly immersed equipment may be increased by the addition of forced circulation of the coolant. However, this additional cooling is at the expense of more power to operate the pump and the additional equipment. The weight of directly immersed equipment may be reduced somewhat by spraying the coolant over the heat producing parts and collecting the heat bearing coolant in the bottom of the container and then pumping it through a heat exchanger and back to the spray nozzles. Such a cooling system represents a saving in the amount of coolant liquid required, but requires a higher pressure pump and consequently more power to run the pump than in the case of the completely immersed equipment.

Liquid cooling is most applicable to power supplies, modulators, servo amplifiers and wide band low frequency amplifiers. It can also be used with certain radio frequency circuits, if consideration is given to the dielectric constant and dissipation factor of the fluid. The coolant must be chemically and electrically compatible with the electronic parts and the case. If liquid cooling is applied to equipments which operate over a wide range of environmental temperatures, care must be exercised in making sure that the coolant can not freeze at the lower temperatures.

Liquid cooling frequently permits a greater degree of miniaturization because of the larger permissable heat concentrations. Further, if a coolant with a high dielectric strength is used, voltage ratings can be increased. On the negative side, liquid cooling requires that the containers accomodate expansion at elevated temperatures. Unless the coolant is chemically inert, it may decompose the electronic parts. Also, maintenance difficulties are increased, and a leak may disable the unit. Repairing of direct liquid cooled equipment is complicated by the necessity for draining the fluid from the unit before working on it. Further, the fluid may be contaminated when the unit is unsealed unless extreme care is used.

5. Direct Vaporization Cooling

Vaporization cooling is the most effective heat removal method known. It has the advantages and disadvantages of direct liquid cooling together with greatly increased cooling. (See Fig. 50) Expendable systems are simple, but involve disposal of the vapor and replacement of the coolant. Non-expendable or continuous systems are complex, expensive, and necessitate the use of a heat exchanger to condense the vapor back into a fluid. Vaporization cooling systems are particularly suited to installations with extremely high heat concentrations and those installations wherein no sink is available or the sink is remotely located. This is discussed further in Vaporization Cooling Section VIII.

C. HEAT TRANSFER TO THE ULTIMATE SINK

1. General

The method of transfer of heat from the subassembly or unit chassis to the sink is dependent upon the method of heat removal from within the subassembly due to the common connection between the two phases of heat rejection. Further, the selection of the optimum method of heat transfer for use in this phase is dependent upon the type of sink available, its location, and its temperature. The sink temperatures, both before and after installation of the equipment, must be considered, since the temperature of local or intermediate sinks may increase when the additional heat is added.

2. Natural Methods

Natural heat transfer from miniaturized subassemblies to the intermediate sink is probably best accomplished by metallic conduction cooling. In general, the reasoning discussed in part B.2 of this section is also applicable. However, the intermediate sink cannot be located at any significant distance from the subassemblies. Small temperature gradients are only obtained over appreciable distances with metallic conduction cooling when large heat conductors are used. The cost and weight of such conductors will probably be excessive. In certain instances structural parts may be used, i.e. the equipment may be thermally fastened to the hull of a ship.

Natural convection and radiation may be used at the sink if the sink is air of a relatively low temperature. The maximum heat dissipated by the surfaces should seldom exceed 0.25 watts per sq. in. and should be limited to approximately 0.50 watts per sq. in. Even so, relatively high temperatures can easily be achieved. It is therefore recommended that this mode of cooling be used only with equipments of low heat concentration, provided that the rejected heat is not introduced into other nearby equipment.

3. Forced Air

Forced air is more applicable to this phase of cooling than natural methods, particularly if the sink is nearby air. The air should be properly directed and distributed over the sub-assemblies. Unit heat dissipations of the order of 2 watts per sq. in. can be obtained readily. Supplemental data are presented in Forced Air Section VI.

4. Indirect Liquid Cooling

When electronic equipment is to be operated in high temperature environments at high heat concentrations or when the sink is located at a distance from the equipment, optimum cooling can be achieved by a forced liquid cooling system. Greatly increased cooling over that obtained by forced air is possible. This cooling mode, using fresh water, is especially recommended for shipboard usage. Almost any reasonable degree of cooling can be attained through the use of cold plate or cold panel heat exchangers. See Sec. VII F.

Indirect liquid cooling systems utilizing continuously circulated fresh water appear to be especially applicable to shipboard electronic equipment. The heat can be removed from the fresh water in a fresh water to sea water heat exchanger. Work on such systems is continuing at this Laboratory.

5. Indirect Vaporization Cooling

This mode of heat transfer will provide the maximum obtainable cooling. It is recommended for use only with devices having extremely high heat concentrations. Its general application to miniaturized equipment remains to be determined.

D. DESIGN EXAMPLE OF THE SELECTION OF OPTIMUM COOLING METHODS - Example (8)

1. The Problem:

Construction of a piece of electronic equipment which dissipates 300 watts is contemplated. It is planned to package it in a cabinet 9.75 in. x 15 in. x 17 in., which is to be located in air at a normal room temperature.

a. Will any special cooling considerations be required for this package?

b. Can this package be made smaller?

$$\text{Heat concentration} = \frac{\text{Dissipated power}}{\text{Volume}}$$

$$= \frac{300}{9.75 \times 15 \times 17} = \frac{300}{2486}$$

$$= .12 \text{ watts/cu.in.}$$

This is a low heat concentration. No particular cooling considerations are required provided that the unit heat dissipation is adequate. See Fig. 50.

$$\text{Unit heat dissipation} = \frac{\text{Dissipated power}}{\text{Area of cooling surface}}$$

$$= \frac{300}{2 \times 9.75 \times 17 + 2 \times 9.75 \times 15 + 2 \times 15 \times 7} = \frac{300}{1135}$$

$$= .26 \text{ watts/sq.in.}$$

Referring to Fig. 50, note that the maximum unit dissipation for free air cooled surfaces is in the neighborhood of 0.25 watts/sq.in. Thus, the package with .26 watts/sq.in. surface area will be satisfactory and no special cooling means will be required.

From Fig. 50 note that a unit heat dissipation of one watt/sq. in. of surface area may be feasible if, for example, forced air cooling is used on the external surfaces. Thus, with forced air cooling it may be possible to miniaturize the package from a surface area of 1135 sq. in. to a surface area of 300 sq. in. or less, dependent upon the Reynolds number and provided that the heat concentration is not excessive.

External dimensions of 7 in. x 5 in. x 10 in. appear in order for the miniaturized unit.

$$\text{Unit heat dissipation} = \frac{300}{7 \times 5 \times 2 + 5 \times 10 \times 2 + 7 \times 10 \times 2}$$

$$= \frac{300}{310} = \text{approx. 1 watt/sq.in.}$$

$$\text{Heat concentration} = \frac{300}{7 \times 5 \times 10} = \frac{300}{350}$$

$$= .85 \text{ watts/cu.in.}$$

This is a fairly high heat concentration. Metallic conduction cooling could be used within the unit satisfactorily if paths of low thermal resistance to the external surfaces are incorporated. Liquid potting could be used as an alternate technique. (See Fig. 50)

X. THE TEMPERATURE LIMITS OF ELECTRONIC PARTS

A. GENERAL

One of the primary problems facing the electronic designer in the thermal design of electronic equipment is the determination of the maximum temperatures which electronic parts can withstand. The maximum temperature is usually limited by the thermal coefficients of the electrical characteristics of the part and their effect on electronic performance, the degree of reliability and life desired, and the temperature which the part can survive without outright failure. The thermal effects on electronic performance are considered to be peculiar to the circuits involved and indirectly related to the heat removal problem. Therefore, a discussion on the thermal coefficients of electrical parameters is not included in this manual. It is realized that the degree of cooling can alter the electronic performance by lowering the temperature spread and, in some instances, special cooling means will be required for this purpose. However, in most electronic equipment, the life, reliability and survival temperature are the primary thermal factors.

Much remains to be accomplished in the determination of temperature vs. life or reliability for electronic parts. In general, such information is difficult to obtain. A reasonable collection of material on the maximum survival temperatures of parts is available. However, most of this data is in terms of ambient temperature which is inadequate in the design of densely packaged electronic equipment. It is anticipated that the parts manufacturers will ultimately rerate their products and also assist in determining their compatibility with liquid coolants.

B. THE THERMAL LIMITATIONS OF VACUUM TUBES

1. General

Vacuum tubes are considered by some electronic engineers to be less temperature sensitive than most electronic parts. This is partially correct. The electronic parameters of tubes are stable within wide temperature limits for certain life periods. However, if tubes are operated beyond safe temperature limits their life and electrical characteristics will be significantly curtailed in a relatively short period. Further, the removal of heat from vacuum tubes is extremely important, since they are usually the primary heat sources in equipment. Overheating of vacuum tubes can lead to shortened tube life through: the accelerated formation of gas, resulting in positive shifts in bias and progressive loss of emission; the thermal expansion of internal parts, causing shorts and changes in tube characteristics; the formation of leakage paths, especially heater to cathode leakage; changed contact potential; the formation of mechanical stresses in glass resulting in envelope failures; and the accelerated development of cathode interface resistance. It is also desirable to operate tubes as cool as possible to avoid mechanical failures due to creep and fatigue of metals, particularly when the tubes are subjected to impact and vibration.

In general, vacuum tubes are properly rated in that "hot spot" bulb temperature ratings have been assigned by the manufacturers. Certain tube groups have similar temperature ratings, for example, the premium subminiature tubes. Most of the special tubes, ruggedized tubes and JAN tubes have specific thermal ratings. The conventional receiving type tubes do not appear to be completely provided with such ratings. In fact, some conventional tubes have envelope glass which is different from that of others of the same type. It is recommended that the tube manufacturers' ratings be closely followed. Most manufacturers welcome requests for such information.

2. Heat Transfer Within Vacuum Tubes

The modes of heat removal within a vacuum tube are complex. A high temperature emitting surface is necessary to maintain proper electronic emission. Heater temperatures range from 1000° to 1300°C and cathodes operate in the neighborhood of 750°C. Tube structures are designed so that the thermal resistance from the heater and cathode to the envelope and leads is as great as possible in order to reduce the heater power to a minimum. However, for circuit purposes, most tubes are provided with low inductance leads to their internal elements. These leads can conduct heat from the cathode and a compromise between these two requirements must be made by vacuum tube manufacturers.

Much of the heat dissipated in vacuum tubes appears at the plate. Almost all the heat produced at the filament, cathode, control grid and screen grid is transmitted by radiation through the vacuum into the plate. The remainder of the heat produced by tube elements, other than the plate, is radiated to the inside surface of the tube envelope and/or conducted into the tube pins by metallic conduction along the tube element leads. The plate is heated not only by the heat received from the other elements but also by its normal dissipated energy. Plate temperatures in vacuum tubes, other than transmitting types, normally range from 350° to 400°C. Almost all the energy dissipated by the plate is transmitted by radiation through the vacuum and is absorbed by the glass envelope. Due to its transmission characteristics, glass begins to be a poor transmitter of infrared radiation at 2.5 microns. Thus, glass is essentially opaque to radiation from sources near 400°C, and only 6% of the energy radiated from the plate is transmitted directly through the glass envelope. The remaining 94% of the heat radiated from the plate is absorbed by the glass. The glass is heated and reradiates part of this energy at a lower temperature level and convects or conducts the remainder to the environment. Some heat from the plate is conducted along the plate lead through the tube pins. When plates operate at temperatures of the order of 750 to 850°C (cherry red), as in tantalum element transmitting tubes, the majority of radiation from the plate passes directly through the glass.

These modes of heat rejection from within a vacuum tube result in a concentration of heat in the glass envelope adjacent to the plate and to some extent at the base of the tube. If a tube is mounted vertically and operated in free air, a small hot spot, due to conduction through the leads, will appear at the base and the envelope will have a definite hot spot at approximately two thirds its height, opposite the plate, due to radiation through the vacuum (see Fig. 51). Glass is a relatively poor heat conductor and marked temperature gradients will appear in the envelope adjacent to the upper and lower edges of the plate structure. It is desirable to cool vacuum tubes in a manner that will reduce such gradients in the envelope. Large temperature differences can cause severe mechanical strains which lead to envelope breakage.

It can be concluded that vacuum tubes must be cooled primarily by removing the heat from the envelopes. A portion of the heat can be removed through the pins or leads at the base. A study of the magnitude of heat removal which can be obtained by tube pin cooling is in work at this Laboratory.

3. Limiting Factors

   a. The release of gas is greatly accelerated when the temperatures of tube elements exceed certain limits. Most vacuum tube elements are degassed at temperatures in the neighborhood of 500°C. The envelopes are usually near 300°C during this process. If the glass or element temperatures are permitted to exceed these values after the tube has been placed in service, gas will be rapidly released, the getter will be unable to absorb the gases, and a gassy tube will result.

   b. Minute amounts of emitting material usually migrate from the cathode to the control grid. Should the grid temperature become excessive, grid emission from these materials can occur.

   c. The glass envelopes of most tubes should not be permitted to operate at temperatures greater than 200°C maximum. Premium tubes are usually provided with a special glass which can be operated at 260°C maximum. The voltages applied to the tube elements can produce electrostatic stresses in the glass at the base. The leakage resistance of glass at high temperatures is much less than at normal temperature and appreciable current flow in the glass is possible.

Fig 51 A
MINIATURE TUBE PAINTED
WITH [illegible]

Fig 51 B
MINIATURE TUBE PAINTED
WITH "THERMOCOLOR" NO 30

4. Vacuum Tube Ratings

In general, the envelope temperature of small receiving type tubes should be held below 175°C "hot spot temperature" for reliable service. A reduction of bulb temperature of from 200°C to 160°C will result in a substantial increase of life expectancy of the tubes. The cooling of the envelope is the most important consideration when mounting the tube (see Section V). The development of miniature and subminiature tubes has led to a large reduction in envelope surface area and a large increase in the rated unit heat dissipation. This means that the maximum envelope temperatures were increased greatly over those of the conventional tube types. Table 12 is representative for several bulb types:

TABLE 12

Unit Heat Dissipations of Typical Tube Types

Bulb Type **	Octal T-9	Miniature T-5 1/2	Subminiature T-3
Bulb area, sq. in.	10.5	4.1	1.7
Maximum dissipation of tube in watts	18.7	16.8	7.8
Unit heat dissipation, watts/sq.in.	1.78	4.1	4.6
Unit heat dissipation ratio (with respect to T-3)	.44	.85	1
Bulb temperature in free air at 23°C	160°C	225°C*	280°C*

Typical envelope temperatures for sea level and at 23°C ambient temperature conditions are presented in Table 13 as an indication of the temperatures obtained with a single tube in free air.

* Note - excessive - not recommended by the author.
** Note - these heat concentrations apply to all of the tube sizes mentioned.

TABLE 13

Typical Bulb Temperatures
at 23°C Ambient

Tube Type	Bulb Size	Percent Maximum Plate Dissipation 20	40	60	80	100
12AU7	T-6 1/2	77°C	100°C	118°C	133°C	146°C
6C4	T-5 1/2	64°C	82°C	98°C	113°C	125°C
6AH6	T-5 1/2	88°C	103°C	116°C	126°C	132°C
5U4G	ST-16	105°C	116°C	127°C	138°C	149°C
5687	T-6 1/2	123°C	140°C	155°C	155°C	183°C

With increased ambient temperature the following envelope hot spot temperatures are reached for the above envelope types:

TABLE 14

Approximate Bulb Temperatures at Various Ambient Temperatures
(General values - not including correction for shapes)

Ambient Temp. (Sea level pressure)	Unit Heat Dissipation in Watts per sq. in. 1.0	2.0	3.0	4.0	5.0
23°C	100°C	170°C	230°C	*280°C	*310°C
160°C	220°C	260°C	*300°C	*340°C	*370°C
250°C	*310°C	*350°C	*390°C	*420°C	*450°C

* Note - Excessive - not recommended by author

Fig. 52 presents the relative temperatures of various tube types. Fig. 53 shows the bulb temperature vs. ambient temperature for types 6AK5 and 5702.

It is apparent that, at maximum ratings, excessive vacuum tube temperatures can be obtained even in free air. When tubes are used in equipment, these free air ambient conditions seldom exist. It is strongly recommended that the bulb temperature of each tube used in an equipment under design or development be measured to make certain that safe operating temperatures are achieved. Cornell Aeronautical Laboratory Report HF-845-D-2 - "Manual of Standard Temperature Measuring Techniques, Units and Terminology" presents methods for such temperature measurements.

CAUTION: Extremely effective cooling can reduce the envelope temperature to a level which is far below the maximum rated temperature. This is an excellent practice but it should not be used to increase the internal element dissipation of tubes beyond their nominal rated values. Such a practice is hazardous and overcooling should not be used in order to exceed the maximum rated power level of any tube.

C. THE THERMAL LIMITATIONS OF SEMICONDUCTOR DEVICES

1. GENERAL

Semiconductor devices are more temperature sensitive than vacuum tubes. In a vacuum tube, the emitter (cathode or filament) operates at a relatively constant temperature which is essentially independent of the environmental temperature. Such is not the case in a semiconductor, since the temperatures of the active elements are directly related to that of the environment. As a result, a change in operating temperature usually modifies the electrical characteristics of the device. Further, most semiconductors malfunction completely and become conductors at elevated temperatures.

Certain circuits tend to stabilize the performance of transistors. Such circuits are not within the scope of this report. In general, it is desirable to operate semiconductors at the lowest practical temperatures. This will permit a minimum of derating and maximum performance. Unfortunately, only ambient temperature ratings are available for most of the semiconductor devices. Several manufacturers are in the process of determining surface and junction temperature ratings for transistors.

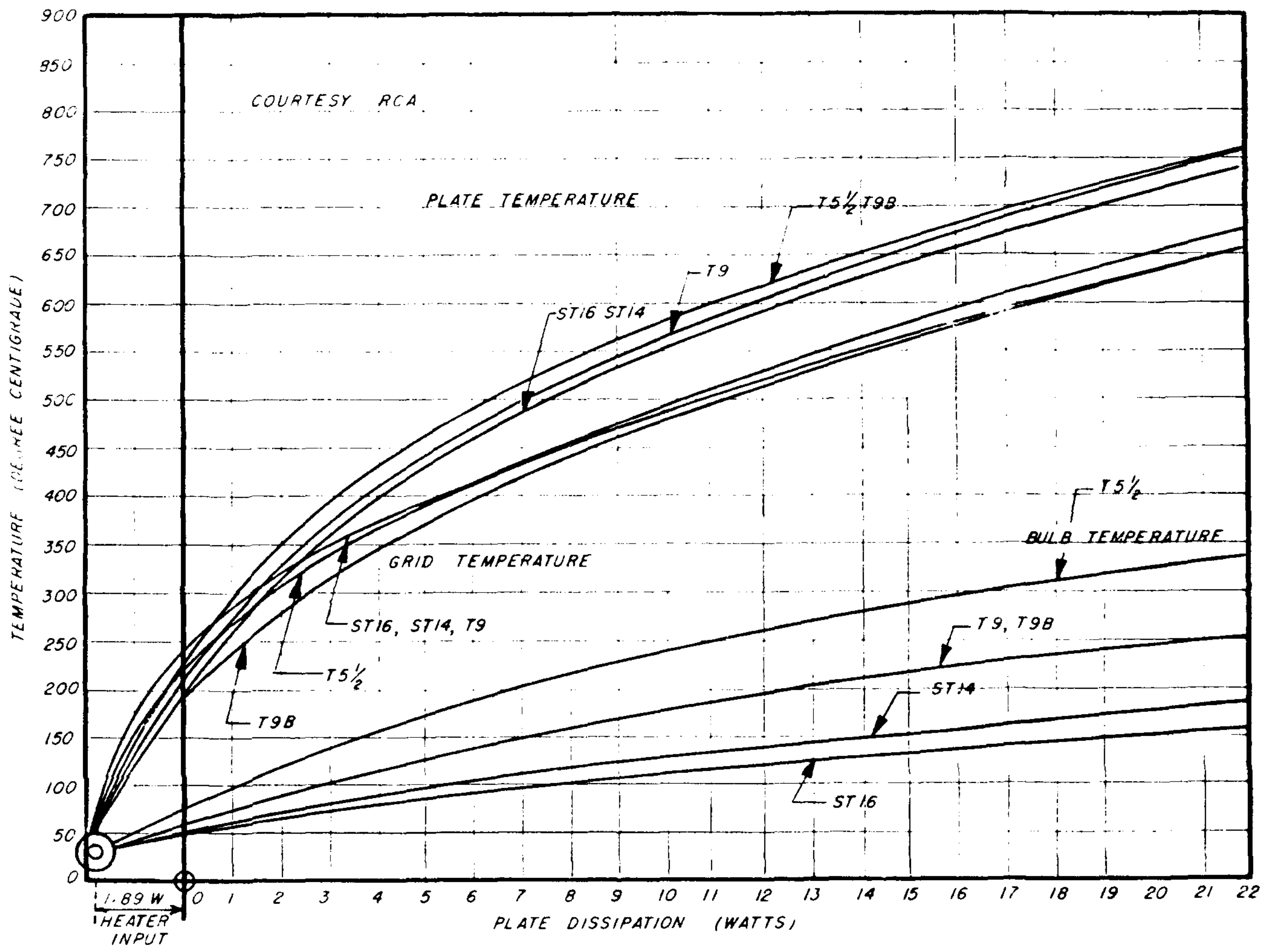


Fig. 52 - RELATIVE TEMPERATURES OF TYPICAL TUBES

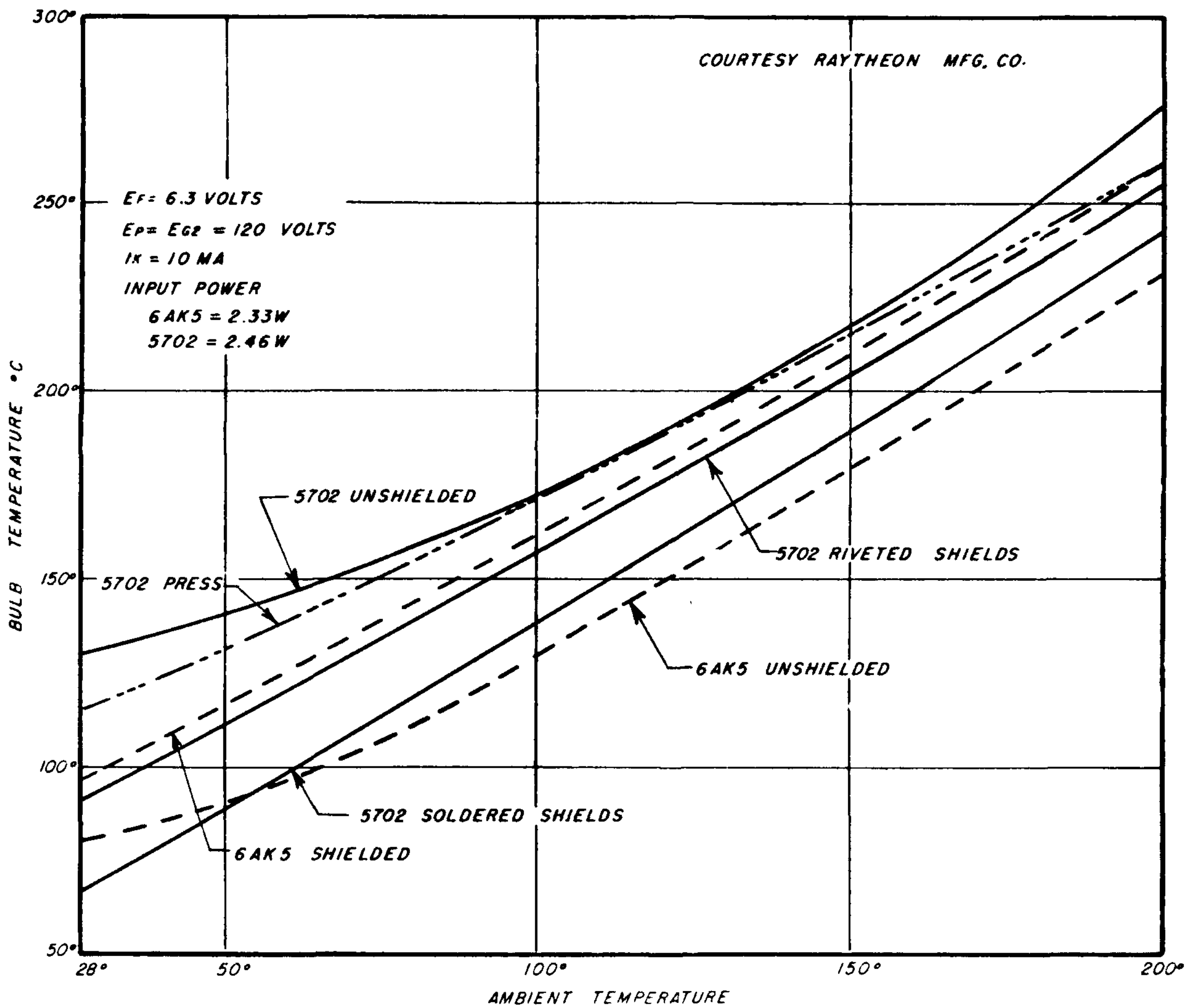


Fig. 53 - BULB TEMPERATURE VS. AMBIENT TEMPERATURE, 6AK5 VS. 5702

2. Selenium Rectifiers

Selenium rectifiers should be derated above 50°C ambient temperature when more than 1000 hours of life are desired. At no time should the peak surface temperature exceed 135°C. The following table displays typical deratings:

Ambient Temp.	Per Cent of Rated Voltage	Per Cent of Rated Current
45°C	100	100
60°C	100 80	80 100
65°C	100 80	65 80
70°C	100 80	50 65
75°C	80 60	45 60
80°C	80 60	30 45
80°C max.	50	40

Fig. 54 also presents derating curves.

3. Germanium Diodes and Transistors

In general, these devices are extremely temperature sensitive. Germanium temperatures of from 85 to 100°C at the junction or point of contact usually result in the loss of the semiconducting characteristics. As soon as the temperatures are reduced, the germanium can recover with only minor ill effects. However, indium and other low melting temperature materials are frequently used in germanium devices and it is possible to permanently damage them by overheating. Usually, it is not recommended that germanium devices be operated at peak internal temperatures exceeding 75°C. Certain germanium and silicon devices for operation at higher temperature levels are now under development. Manufacturers' ratings on these devices must be accurately followed.

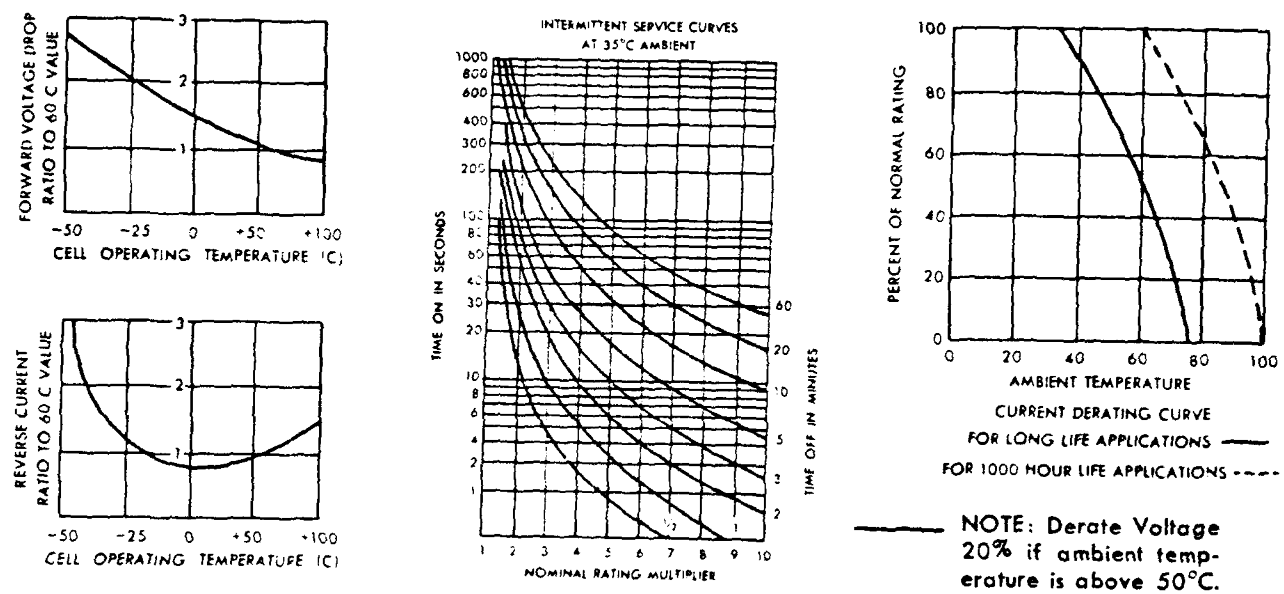


Fig. 54

SELENIUM RECTIFIER RATINGS

4. Magnesium Copper Sulphide Rectifiers

Hermetically sealed rectifier cells of these materials can be used at temperatures as high as 200°C ambient. Typical ratings follow: Operating temperature range - from 70° to 200°C ambient. At ambient temperatures lower than 100°C, operation at full rating should result in a life expectancy of at least 1000 hours. When full life (nominally 10,000 hours) is desired at ambient temperatures between 40° and 100°C, the following derating formula is recommended:

$$\text{Per cent of } 40^{\circ}\text{C D.C. current rating} = 100 \times \left(\frac{130 - \text{max.amb.temp.in } ^{\circ}\text{C}}{90}\right)$$

5. Silicon Diodes

In general, silicon diodes can be used at maximum temperatures ranging from 100 to 110°C. Silicon diodes and transistors for operation at 200°C are under development. Other thermal characteristics are similar to germanium diodes.

6. Copper Oxide Rectifiers

In the region of the limiting value, that is, small current or voltage, the dependence of such rectifiers on temperature is very great. The internal resistance decreases exponentially with increasing temperature. If the load resistance is significantly larger than the internal resistance, the effects of temperature sensitivity can be made to be negligible. Copper oxide rectifiers are rated for full load operation at 40 to 50°C ambient temperature.

7. Dry Disk Titanium Rectifiers

These devices show promise for use at 125°C ambient temperature.

D. THE THERMAL LIMITATIONS OF MAGNETIC CORE DEVICES

1. General

Iron core reactors are considered to be less temperature sensitive, performancewise, than any other electronic part, with the exception of vacuum tubes. When an inductance is used in a frequency controlling element, temperature sensitivity is, of course, present. Iron core reactors are third in order of importance as heat sources. Like vacuum tubes, reactors which are operated beyond safe temperature limits will have short life. Excessive temperatures can cause the failure of insulating materials and conductors leading to high reactor mortality. Insulation does not fail by immediate breakdown upon arrival at some critical temperature, but by gradual mechanical deterioration with time. Ultimately, a short circuit occurs and subsequent "cremation" results. With class A insulation, for example, experience indi-

indicates that the insulation life is halved for each 10 to 12°C increase in temperature throughout the practical operating temperature range. With liquid cooling (oil immersion) the life is halved for each 7 to 10°C increase in temperature.

The heat generated in reactors is produced in the magnetic core and in the conductors. The primary mode of heat transfer within conventional iron core reactors is conduction. Due to the necessity for turn to turn and layer to layer electrical insulation, the thermal resistance between internal hot spots and the surface is large, and high winding temperatures can be obtained. If surface temperature ratings are not available, the internal temperatures of reactors should be determined with imbedded thermocouples.

2. Limiting Insulation Temperature

The life at the limiting temperature for any one class of insulation may vary widely according to the quality of the material used, the construction techniques, the mechanical strains imposed on the insulation, and the kind of service to which it is applied. From the results of experience with equipment in service and from laboratory tests on various insulating materials, limiting insulation temperatures (called hottest spot temperatures) have been assigned by the AIEE. They are of primary importance and useful as a point of reference or "bench mark" in selecting the practical values of observable temperature rise. "Hottest spot" temperature values are not directly applicable for use in rating since the "observable" temperature, that is, the temperature which is directly measurable in practical tests, is less than these peak temperatures by an amount which may be widely different for various types and sizes of reactors. This is due to the inaccessibility of the hottest spot, non-uniformity of cooling, the thermal conductivity and thickness of the insulation, the form of winding, the rate of heat flow and the relative locations of the "hottest spot" and the cooled surfaces. Therefore, temperature difference allowances are included in the "hottest spot" (peak) ratings.

3. Class 0 Reactors

Class 0 insulation consists of cotton, silk, paper and similar organic materials when neither impregnated nor immersed in a liquid dielectric.

The maximum peak temperature for class 0 insulation is 90°C. The temperature difference allowance between the "hottest spot" and the temperature measuring devices is approximately 5°C and the maximum indicated temperature is therefore limited to 85°C.

4. Class A Reactors

Class A insulation, as defined by A.I.E.E., consists of (1) cotton, silk, paper and similar organic materials when impregnated or immersed in a liquid dielectric; (2) molded and laminated materials with cellulose filler, phenolic resins and similar resins; (3) films and sheets of cellulose acetate and other cellulose derivatives of similar properties; and (4) varnishes (enamels) as applied to conductors.

The usual maximum peak temperature for class A insulation is 105°C. The temperature difference allowance between the "hottest spot" and the temperature sensing element is approximately 5°C and maximum indicated temperature is therefore limited to 100°C.

5. Class B Reactors

Class B insulation consists of mica, asbestos, Mylar, Fiberglas and similar inorganic materials in built-up form with organic binding substances. Composite magnet wire insulation consisting of Fiberglas layers covering polyvinyl acetal or polyamide films are included in this class.

The maximum peak temperature for Class B insulation is 130°C. The temperature difference allowance between the "hottest spot" and the temperature sensing element is approximately 10°C and the maximum indicated temperature is therefore limited to 120°C.

6. Class H Reactors

Class H insulation consists of (1) mica, asbestos, Fiberglas and similar inorganic materials in built-up form with binding substances composed of silicone compounds materials with equivalent properties; (2) Teflon, silicone unds or materials with similar properties.

The usual maximum peak temperature for class H insulation is 250 to 275°C. The temperature difference allowance between the "hottest spot" and the temperature sensing element is of the order of 20°C and the maximum indicated temperature is therefore limited to approximately 230°C. For long life it is recommended that the maximum indicated temperature be reduced to 200°C.

7. Class C Reactors

Class C insulation consists entirely of mica, porcelain, glass, quartz and similar inorganic compounds. No upper temperature limits have been selected for this class of insulation. It is anticipated that the limit will be in the neighborhood of 260°C because of the electrolysis which can occur in glasslike materials at temperatures exceeding this level.

E. THE THERMAL LIMITATIONS OF RESISTORS

1. General

Resistors rank second to vacuum tubes in the order of magnitude as heat sources. Almost all resistors have been designed for cooling by natural means. Information related to their ratings when liquid cooled is practically non-existent. In general, resistors are provided with lugs or leads for free space mounting and are thus cooled by conduction through the mountings, together with whatever radiation and convection is present. The lead length and temperature of the points of attachment therefore can greatly influence the operating temperature of a resistor in a given location. One of the factors which limit the maximum temperature of many types of resistors is oxidation. If the protective surface material, which is usually an enamel or varnish, is damaged and exposed to the atmosphere, oxidation of the resistance material occurs rapidly and the resistor is destroyed. Hermetic sealing and inerting will tend to overcome this difficulty. The ratings in the following sections are based primarily on ambient rating in free air. Since the military requirements may be more stringent than the commercial ratings, where different ratings occur, the Bureau of Ships' ratings are included in parenthesis.

2. Fixed Resistors

a. Carbon film resistors

Pyrolytic carbon, borocarbon, cracked carbon and other similar resistors have essentially identical ratings for a given size. They are rated for full wattage dissipation at 40°C ambient and linearily derated to zero wattage at 150°C (120°C) ambient. The maximum "hot spot" surface temperature for reliable service is 120°C to 150°C, dependent upon the allowable change in value.

b. Composition carbon resistors

These resistors are rated for full wattage dissipation at 40°C ambient and linearily derated to zero wattage at 100°C (70°C) ambient. The maximum "hot spot" surface temperatures range from 110 to 130°C.

c. Printed resistors

The ratings of printed resistors vary widely, being dependent upon the formulations utilized and the materials upon which they are printed. In general, printed resistors are rated for full wattage dissipation at 40°C ambient and linearily derated to zero dissipation at 75°C ambient. The maximum "hot spot" surface temperature is usually 85°C.

d. Palladium film resistors

These metallized film resistors are rated for full wattage dissipation at 100°C ambient and linearily derated to zero wattage at 150°C ambient. The maximum "hot spot" surface temperature for long life is 200°C. Under short life conditions "hot spot" surface temperatures as high as 230°C can be tolerated.

e. Glass resistors

Resistors consisting of a conducting metallic oxide film on glass are rated for full wattage dissipation at 40°C ambient and linearily derated to zero wattage at 200°C ambient. The maximum "hot spot" surface temperature for reliable service is 225°C.

f. Molded wire wound resistors

Wire wound resistors which are molded in phenolic materials are rated for full wattage dissipation at 70°C ambient and linearily derated to zero wattage at 150°C ambient. Maximum surface temperatures should not exceed 150°C.

g. Wire wound vitreous enamel resistors

The average wire wound resistor of this class is rated for full wattage dissipation at 40°C (25°C) ambient and derated almost linearly to zero wattage at 225°C (160°C) ambient, dependent upon the type. The maximum surface temperature is usually approximately 250°C (275°). Embedded and similar high temperature wire wound vitreous resistors are rated for full dissipation at 40°C ambient and are essentially derated linearly to zero wattage at 400°C. Their maximum surface temperature is 415°C. Several miniature vitreous wire wound resistors produced in England have the same ratings as the embedded types. Liquid cooled ratings are available from only a few manufacturers.

3. Variable Resistors

a. Wire wound variable resistors

Conventional low power, low operating temperature, wire wound potentiometers with a linear taper are rated from two to four watts maximum dissipation, dependent upon their size, at 40°C ambient and are linearly derated to zero wattage at 105°C (100°C) ambient. Non-linear tapered controls are rated for .01 watt dissipation per degree of rotation at 40°C. The ratings of power rheostats vary, dependent upon their constituent materials.

b. Composition carbon variable resistors

Ordinary variable carbon resistors are rated for one half watt maximum dissipation at 40°C, linearly derated to zero at 75°C (85°C). Special high temperature units are rated from two to one half watt maximum dissipation, dependent upon their size, at 70°C, and are linearly derated to zero at 150°C (120°C).

F. THE THERMAL LIMITATIONS OF CAPACITORS

1. General

Capacitors are not normally considered to be heat sources, with the exception of electrolytic capacitors having high leakage currents and capacitors with relatively high loss factors in radio frequency circuits in transmitters. The surface temperatures of capacitors are usually those of the thermal environment. In general, the leakage resistance of capacitors decreases with temperature, so that their usable maximum temperature is determined by the permissible circuit losses and their survival temperatures.

2. Fixed Capacitors

a. Paper dielectric capacitors

Most paper capacitors have an upper temperature limit of 75 to 85°C. High temperature paper capacitors have an upper temperature limit of 125°C.

b. Synthetic film dielectric capacitors

Capacitors using "Teflon", "Mylar", "Thermofilm" and similar plastic dielectrics show promise as substitutes for paper dielectric capacitors at temperatures ranging from 130 to 200°C ambient. Such capacitors are still under development and until more definite information relative to the life and thermal capabilities of these capacitors are known, it is suggested that the application recommendations of each manufacturer be closely followed.

c. Glass dielectric capacitors

Glass capacitors are rated for service at 200°C maximum.

d. Mica dielectric capacitors

Mica dielectric capacitors are limited to peak temperatures of the order of 120°C when they have plastic cases. Mica is an excellent high temperature dielectric. Mica capacitors with metal cases, or without cases, can be used at elevated temperatures.

e. Vitreous enamel capacitors

These capacitors are rated for 200°C service.

f. Barium titanate dielectric capacitors

Capacitors with high K ceramic dielectrics have upper temperature limits of the order of 85°C.

g. Electrolytic capacitors

High quality conventional electrolytic capacitors are rated to 85°C maximum ambient temperature. Tantalum electrolytic capacitors, dependent upon the type, are rated at 125, 150, 175 and 200°C maximum ambient temperatures, respectively.

3. Variable Capacitors

Almost all variable capacitors, with the exception of the barium titanate dielectric type, use dielectric materials capable of service at 200°C.

XI. RECOMMENDATIONS FOR DETERMINING AND IMPROVING THE THERMAL PERFORMANCE OF ELECTRONIC EQUIPMENT

A. THE THERMAL ANALYSIS OF ELECTRONIC EQUIPMENT

The adequacy of a given thermal design can be determined by test under either actual or simulated conditions and environments. Such a test may involve many considerations, measurements, and evaluation procedures. Such items as temperature, pressure, and air flow measurements, proper simulation of the design environment, blower power measurements and electronic performance are involved. Caution should be exercised in avoiding radiation and convection effects associated with nearby walls. Also, stray air currents should be eliminated.

Coverage of all these items is beyond the scope of this manual. However, excellent treatment of air-cooled equipment is contained in AF Technical Report No. 6579 (see Ref. 6). This report is a comprehensive manual on the testing and thermal evaluation of air-cooled electronic equipment. The factors outlined in Section III pertaining to environmental ratings should be considered. In general, the temperature rise within an equipment will be relatively constant over a fair portion of the operating temperature range, all other factors being unchanged. If modifications to the cooling system are necessary, all thermal tests should be repeated after modification.

B. IMPROVING THE THERMAL PERFORMANCE OF EXISTING ELECTRONIC EQUIPMENT

Within certain limits the thermal performance of existing equipment can be improved by modification. Improved natural cooling methods and other similar simple techniques can be incorporated into much of the current equipment.

The variety of cooling methods for existing electronic equipment which is either malfunctioning due to heat or which must be operated in thermal environments more severe than originally specified is limited. An optimum cooling method for such equipment would probably depend on the complexity of the installation, space, power, location, cooling media available, cost, and other such considerations.

In general, the addition of forced convection may be the most practical method. Another possible method is to cool the equipment cabinet with a water cooled heat exchanger in intimate contact with the surface of the case. Shipboard electronic equipment that has been designed for cooling by forced air and is mounted in relay racks and cabinets can be modified by installing water cooled heat exchangers (cold panels) on the inside surface of the cabinet. The air inside the cabinet should be recirculated through the equipment and across the cold panels.

Consider an electronic equipment which operates satisfactorily in a specified thermal environment but must now function in a compartment with a higher temperature environment. Malfunction will probably result if supplemental cooling means are not provided. Several pertinent considerations follow:

1. Can the equipment operate satisfactorily using compartment air in forced circulation within or around the equipment? If it appears that compartment air can be used, other factors, including compartment air inlet and outlet facilities, air ducting requirements and fan power requirements, should be considered. It must be remembered that the rejected heat in the form of hot air cannot, in general, be dissipated into a compartment without increasing the temperature within the compartment.

   Existing ventilation and air conditioning systems cannot always be utilized for equipment cooling. The system may be overloaded by the added heat load, rendering the space uninhabitable for personnel.

   Hot spots within the equipment can be a source of trouble. Air may be forced through ducts and directed upon such hot spots. This is one of the most simple and effective means of hot spot cooling of existing equipment. Care must be exercised to prevent the resulting hot air from overheating other electronic parts which may already be operating at or near their maximum temperatures. Thus, it is important to consider not only inlet air ducting, but the collection and removal of the hot air as well.

   If forced air cooling exists in the equipment, improved air flow distribution within the equipment may be obtained by judicious placement of ducts and baffles. In some instances blowers are located haphazardly. The air should be directed to the temperature limited parts and hot spots. Temperature indicating paints can be applied to locate critical items. Fins may be added where necessary to provide larger cooling surfaces. Where required, critical parts may be relocated in the most advantageous position with respect to the cooling air. Metallic conduction cooling of temperature limited parts can be improved by better mounting methods. See Section V.

2. Will the electronic equipment be placed near other heat sources such as a hot pipe or other equipment? There may be alternative locations available. Also, perhaps radiation shields might be used to minimize heat transfer from other heat sources to the electronic equipment in question.

3. Is water available in the compartment for water cooling of the electronic equipment? For example, water cooled tubing might be soldered to the cabinet. In this instance, make sure that the surfaces involved have sufficient area to remove the heat from

the cabinet. Another possibility is to force the internal air over water cooled extended surface coils. See Section VII. The air could be recirculated in a closed duct system. Alternately, the assemblies and subassemblies can be made into more effective heat exchangers. The electronic equipment may be packaged in an electronic equipment console constructed so as to contain trays with liquid or air to carry the heat away. As previously mentioned, a liquid would be preferable.

4. If sufficient air or water is unavailable for cooling purposes, other means must be considered. One possibility is to force air over cooling coils, the air being recirculated in a closed system. The cooling could be accomplished by a vapor-compression system using a refrigerant such as one of the Freons. The refrigerant condenser can be located remotely for air or sea water cooling. It should be noted however, that considerable equipment is required, including piping, a compressor, a drive motor, a condenser, and the various controls. Further, the work of compression together with the heat removed from the electronic equipment must be removed in the refrigerant condenser. Expendable evaporative cooling could also be used when cooling air or water are not available.

While the above generally recommends ducting air directly to the inside of equipment, it should be pointed out that great care must be taken in the design of the cooling air attachment fittings so that a minimum of space, weight and maintenance time is expended through their usage. The equipment should be easily connected to the duct when the equipment is slid into place on its rack. For simplicity, it is desirable that only air inlet ducting be provided.

The current practice of circulating cooling air at random through a cabinet containing several items of equipment can be very inefficient. With this arrangement, some pieces of equipment may be over cooled while other items may be inadequately cooled. A more efficient cooling system is possible if the cooling air is ducted directly to each equipment item in the quantity required.

C. THE DESIGN OF EFFICIENT ELECTRONIC CIRCUITS

It is well known that the electrical efficiency of most electronic circuits is very low, ranging from almost zero to a few percent. Careful design will, in certain instances, increase the efficiency and help to alleviate the heat removal problem by reducing the dissipated power. It is recommended that each circuit be reanalyzed during its development and if necessary, be redesigned to obtain the highest practical efficiency prior to initiating the heat removal design.

To design efficient electronic circuitry it is necessary to examine each electronic stage individually for useless power being dissipated as heat during stand-by and full-output conditions. Secondly, an analysis of the normal operational duty-cycle should be made. This will provide an indication of where the greatest quantity of power is being wasted and which of the efficiency measures described in this section should be utilized. Fig. 55 presents relative circuit powers.

1. Reduction of Operating Voltages

   In most circuits, excepting those wherein large output signals are required, vacuum tubes will function satisfactorily with adequate gain at plate supply voltages ranging from 80 to 125 volts. It is suggested that plate voltages of this order be applied to reduce the plate dissipation, instead of the usual 200 to 300 volts. In general, the gain will not be significantly reduced and the life will be increased. Further, lower voltage parts which are smaller, lighter, and more economical, can be used.

2. Reduction of Power Dissipated During Stand-By Conditions

   With equipments that have long stand-by periods, as compared to their operational periods in a normal duty-cycle, it is often possible to redesign the equipment to make use of plate circuit switching so that the plate supply will be de-energized during the standby period. Further, it is possible to reduce the filament voltage of vacuum tubes from, for example, 6.3 volts to 6.0 or even 5.9 volts during stand-by without serious effect on the operation of the circuit or the life of the tube.

3. Use of Semiconductor Devices

   Semiconductor devices are more efficient electrically than vacuum tubes. There are instances wherein the application of such a device will serve to increase the overall efficiency of the system. These devices are extremely temperature sensitive and must be treated accordingly. However, with proper care and stable circuit design it is possible to obtain satisfactory performance, especially in relatively low temperature environments.

4. Use of Magnetic Amplifiers

   Electronic engineers are, to some extent, prone to overlook the advantages of magnetic amplifiers. In certain low frequency circuits, saturable reactors have outstanding advantages. The inherent efficiency of a magnetic amplifier is high. Vacuum tubes are essentially variable resistors, dissipating much of their power input. Magnetic amplifiers are variable reactors exhibiting an impedance rather than a resistance. The power loss is correspondingly low.

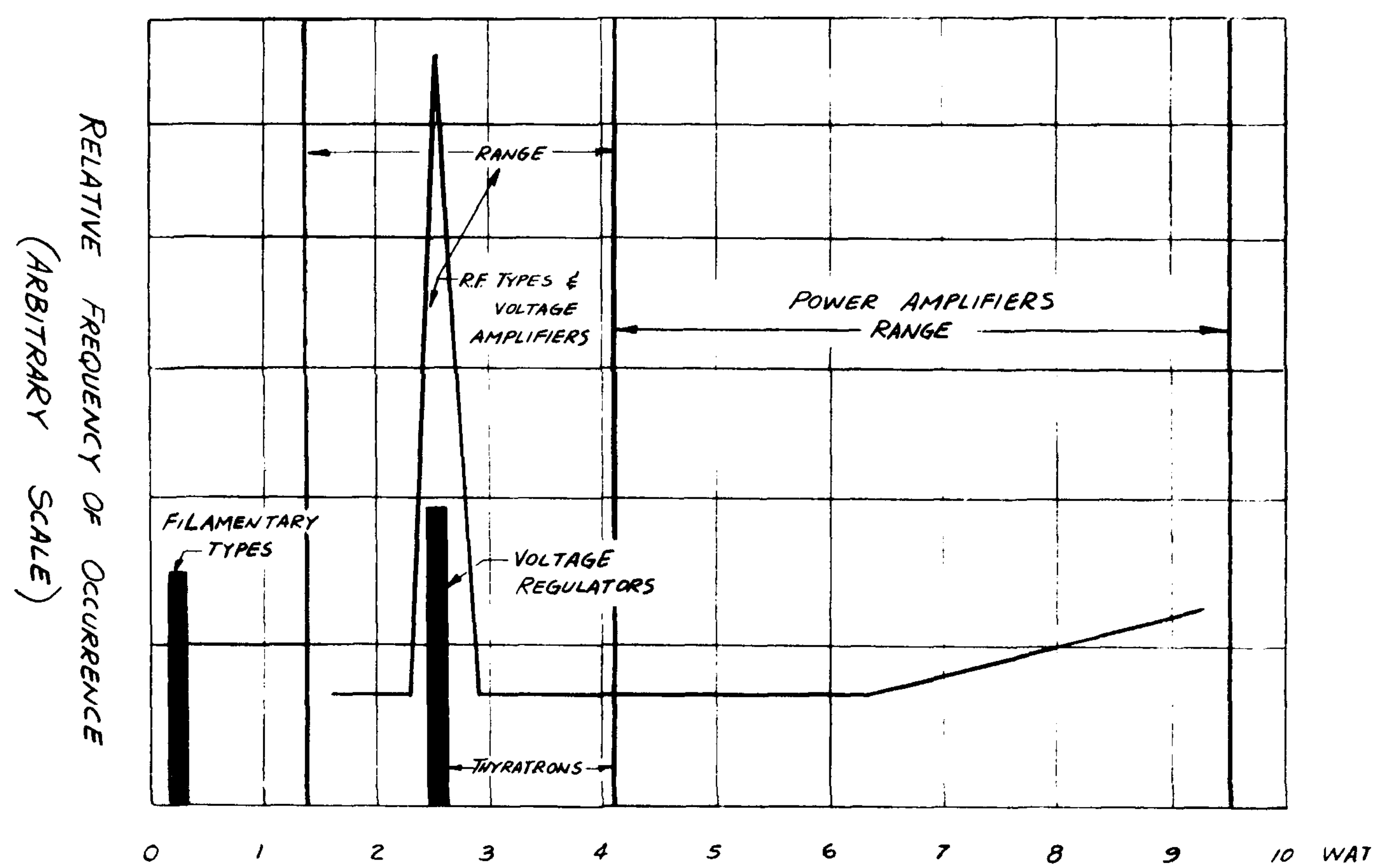
5. Use of Special Circuits

   An examination of most equipments will reveal that only the power output stages require large input power. Often the preceeding stages are operated at excessive power levels with respect to

FIG. 55

THERMAL POWER OF SUB-MINIATURE TUBE CIRCUITS

(BASED ON MAXIMUM TUBE RATINGS)



their function. More efficient designs will decrease the power dissipation and perhaps eliminate some of the intermediate stages. As an example, a radio receiver could operate with the RF and IF amplifiers at low voltages. Only the audio output stage would have high voltage and it could operate in class B. Power output stages should be built to a design limit instead of using, for example, a ten watt stage where only a five watt stage is required. In certain applications semiconductor rectifiers can be used in voltage multiplying circuits for high voltage power supplies, thus eliminating a vacuum tube and power transformer.

In some amplifier applications it is possible to use a pentode tube in a "starved circuit" and achieve an increase in over-all gain together with a reduction in dissipated power. The additional gain may be utilized to eliminate a stage of amplification or be applied as increased feedback to stabilize the system.

In operating a direct-coupled pentode amplifier in a "starved circuit", it is necessary to operate the pentode with a screen voltage below 10 per cent of its plate supply voltage and to increase the plate load resistor to ten or more times its normal value. The starved condition decreases the transconductance and increases the plate resistance. This results in an overall increase of the amplification factor. Additional advantages are that it requires few parts and has a gain of 1000 compared to a gain of 350 in the conventional R-C-coupled circuit. On the negative side, it is limited in high frequency response by the large load resistance in the plate circuit. This deficiency may be overcome to a small degree by the addition of negative feedback in the circuit.

APPENDIX A

Symbols and Nomenclature

Symbol	Quantity	Units Engineering System	Units Commonly Used in Electronics
A	Area, surface or cross-sectional	sq. ft.	sq. in.
AHP	Air horse power	h.p.	
a	Convection modulus equal to $\frac{g\beta\rho^2 c}{\mu k}$	$\frac{1}{(\text{ft.})^3(^\circ\text{F})}$	
b	Constant in convection equations		
C	Constant in convection equations		
c	Specific heat	$\frac{\text{Btu}}{(\text{ft.})(^\circ\text{F})}$	
CFM	Flow rate by volume	$\frac{\text{cu.ft.}}{\text{min.}}$	
D	Diameter, D_e equivalent diameter	ft.	in.
$\frac{dt}{dL}$	Rate of change of temperature with respect to distance	$\frac{^\circ\text{F}}{\text{ft.}}$	$\frac{^\circ\text{C}}{\text{in.}}$
E	Electromotive force		volts
F_a	Configuration factor in radiation equation		
F_e	Emissivity factor in radiation equation		
FHP	Fan horse power	h.p.	
G	Mass velocity	$\frac{\text{lbs.}}{(\text{sq.ft.})(\text{hr.})}$	
g	Acceleration due to gravity	ft./hr.^{2}	
Gr	Grashof number equal to $\frac{g\beta\Delta t L^3\rho^2}{\mu^2}$		

APPENDIX A (Contd.)

Symbols and Nomenclature

Symbol	Quantity	Units Engineering System	Units Commonly Used in Electronics
h	Coefficient of heat transfer	$\frac{\text{Btu}}{(\text{hr.})(\text{sq.ft.})(^{\circ}\text{F})}$	$\frac{\text{watts}}{(\text{sq.in.})(^{\circ}\text{C})}$
h_c	Convection coefficient		
h_r	Radiation coefficient		
$h_{contact}$	Contact coefficient		
I	Current		amperes
k	Thermal conductivity	$\frac{(\text{Btu})(\text{ft.})}{(\text{hr.})(\text{sq.ft.})(^{\circ}\text{F})}$	$\frac{(\text{watts})(\text{in.})}{(\text{sq.in.})(^{\circ}\text{C})}$
L	Length, characteristic length in convection equations	ft.	in.
m	Exponent in convection equation		
N	Number of rows deep in tube banks		
n	Exponent in convection equation		
N_u	Nusselt number equal to $\frac{hL}{k}$		
p	Absolute pressure	$\frac{\text{lbs.}}{\text{sq.in.}}$	
Pr	Prandtl number equal to $\frac{c\mu}{k}$		
q	Rate of heat transfer	$\frac{\text{Btu}}{\text{hr.}}$	watts
q_c	By convection		
q_r	By radiation		
q_T	Total		

APPENDIX A (Contd.)

Symbols and Nomenclature

Symbol	Quantity	Engineering System	Units Commonly Used in Electronics
R	Resistance		
	Electrical resistance		ohms
	Thermal resistance	$\frac{(hr.)(sq.ft.)(^{o}F)}{Btu}$	$\frac{(sq.ft.)(^{o}C)}{watt}$
r	Radius	ft.	in.
Re	Reynolds' number equal to $\frac{LV\rho}{\mu}$		
S_L	Longitudinal pitch of tube banks	ft.	
S_T	Transverse pitch of tube banks	ft.	
T	Absolute temperature	^{o}R	^{o}K
t	Temperature, Δ temperature difference, t_m mean temperature difference	^{o}F	^{o}C
V	Velocity	$\frac{ft.}{hr.}$, $\frac{ft.}{min.}$	
w	Flow rate by mass	$\frac{lbs.}{hr.}$	
β	Coefficient of thermal expansion (for gases numerically equal to reciprocal of absolute temperature)	$\frac{cu.ft.}{(cu.ft.)(^{o}F)}$	
Δ	Difference		
ϵ	Emissivity		
μ	Viscosity	$\frac{lbs.}{(ft.)(hr.)}$	
ρ	Density	$\frac{lbs.}{cu.ft.}$	
Σ	Sum of		
σ	Stephan - Boltzmann constant	$\frac{Btu}{(hr.)(sq.ft.)(^{o}R)^4}$	$\frac{watts}{(sq.in.)(^{o}K)^4}$

APPENDIX B

List of Associated Cornell Aeronautical Laboratory Reports

Description	Report Number	Date of Issue	Classification	Bu Ships Contract No.
Survey Report of the State of the Art of Heat Transfer in Miniaturized Electronic Equipment	HF-710-D-10	3 Mar. 1952	Unclassified	NObsr-49228
Supplemental Bibliography to Survey Report of the State of the Art of Heat Transfer in Miniaturized Electronic Equipment	HF-710-D-10A	3 Mar. 1952	Confidential	NObsr-49228
Manual of Standard Temperature Measuring Techniques, Units, and Terminology for Miniaturized Electronic Equipment	HF-845-D-2	1 June 1953	Unclassified	NObsr-63043
Final Development Report on Standard Packaged Electronic Video Amplifier and Hydrophone Audio Frequency Amplifier Subassemblies	UM-647-D-22	1 Sept. 1953	Unclassified	NObsr-43431
Design Manual of Natural Methods of Cooling Electronic Equipment	HF-845-D-8	Scheduled for Sept. 1954	Unclassified	NObsr-63043
Design Manual of Methods of Liquid Cooling Electronic Equipment	HF-845-D-9	Scheduled for Mar. 1955	Unclassified	NObsr-63043

APPENDIX C TABLES AND CHARTS

TABLE 15

Heat Conduction Data for Various Materials at Approximately 65°C

	Density lb/cu.in.	Heat Conductivity - k watts/(sq.in.)(°C)/in.	BTU/(hr.)(sq.ft.)°F/ft	Comparison with yellow brass on wt. basis*
Silver	0.380	10.6	241	3.6
Copper	0.322	9.7	220	3.9
Gold	0.696	7.5	171	1.4
Aluminum, pure	0.098	5.5	125	7.2
Aluminum, 63S	0.100	5.1	116	6.6
Magnesium	0.063	4.0	91	8.2
High-beryllia ceramics	0.109 to 0.136	1.7 to 3.9	38.7 to 88.7	1.65 to 5.2
Red brass	0.316	2.8	63.7	1.15
Yellow brass	0.310	2.4	54.6	1.0
Beryllium copper	0.297	2.1	47.8	0.9
Pure iron	0.284	1.9	43.2	0.86
Phosphor bronze	0.318	1.3	29.6	0.53
Soft steel	0.284	1.18	26.8	0.54
Monel	0.318	0.9	20.5	0.36
Lead	0.409	0.83	18.9	0.26
Hard steel	0.284	0.65	14.8	0.30
Steatite	0.094	0.06	13.6	0.08
Pyrex	0.094	0.032	0.728	0.044
Grade A Lava	0.085	0.03	0.683	0.045
Soft glass	0.094	0.025	0.569	0.034
Water	0.0361	0.0167	0.380	0.26
Mica	0.101	0.015	0.341	0.017
Paper-base phenolic	0.0497	0.007	0.159	0.018
Plexiglas	0.043	0.0047	0.107	0.016
P-43 castin resin	0.045	0.0046	0.105	0.103
Maple	0.025	0.0042	0.096	0.022
Pine	0.018	0.003	0.067	0.021
Polystyrene	0.038	0.0027	0.061	0.009
Glass wool	0.001	0.001	0.023	0.13
Air	0.000043	0.0007	0.016	0.21

*Computed as follows:

$$\frac{\text{heat conductivity (material)}}{\text{heat conductivity (yellow brass)}} \times \frac{\text{density (yellow brass)}}{\text{density (material)}}$$

(From Ref. 1)

APPENDIX C (Contd.)

TABLE 16
PHYSICAL PROPERTIES OF THE USEFUL METALS

SUBSTANCE	SPECIFIC GRAVITY	SPECIFIC HEAT	MELTING POINT, DEG. F	BOILING POINT, DEG. F	CUBICAL EXPANSION BY HEAT FROM 32 F TO 212 F	HEAT CONDUCTIVITY, SILVER = 100	ELECTRICAL CONDUCTIVITY SILVER = 100	TENSILE STRENGTH, LB. PER SQ. IN.
Aluminum	2.67	....	1217	3272	0.0070	48	53	18,000
Antimony	6.76	.050	1166	2624	.027 – .050	4.2	3.5	1,000
Bismuth	9.82	.0301	520	2300	.0040	1.8	1.13	6,400
Brass	7.8 – 8.6	.092	1650±	....	.0057 – .0064	15 – 30	23 – 17	9,000 – 40,000
Bronze	8.52 – 8.96	.086	1650±	....	.0051 – .0057	....	....	3,500 – 25,000
Cadmium	8.65	.0567	609.6	1430	.0094	....	....	......
Cobalt	8.55	.107	2696	....	.0037	....	19.9	34,400
Copper	8.85	.093	1981.4	5050	.0051	89	99.5	30,000
German silver	8.5	.095	1850±	....	.0055	8	10 – 32	......
Gold	19.258	.032	1945.5	3992	.0044	53.2	76.7	14,000
Iridium	22.38	.032	4280±	....	.0020	34	30	......
Iron	7.9	.113	2786	4442	.0036	11 – 18	9.9 – 17	39,500±
Iron, cast	7.22	.1298	1900 – 2200	....	.0033	11.9	2.8 – 1.4	......
Iron, wrought	7.70	.1138	2700 – 2900	....	.0035	....	17	50,000±
Lead	11.38	.031	621.3	2900 – 3600	.0088	8.2	7.6	1,600 – 2,400
Magnesium	1.75	.025	1204	2048	.0083	37.6	35.8	20,000
Manganese	8.0	....	2246	3452	.....	....	....	......
Mercury	13.58	.033	-37.97	680	.0182	1.8	1.7	......
Nickel	8.8	.109	2642	....	.0038	14	14.5 – 9.9	50,000 – 100,000
Osmium	22.5	.031	4890±	....	.0020	....	16	......
Palladium	12.0	.058	2822	....	.0036	17	15	50,000
Platinum	21.5	.032	3191	....	.0027	17	20 – 10	30,000 – 50,000
Rhodium	12.4	.058	3542	....	.0026	30	23	......
Silver	10.51	.057	1760.9	3550	.0058	100	100	36,000
Steel	7.9	.117	2550±	....	.0041 – .0030	6 – 14	16 – 3	50,000 – 20,000
Tantalum	14.1 – 16.1	.036	5250	....	.0024	....	9.9	......
Tin	7.35	.056	449.4	3800	.0069	15.2	11.3	5.00
Titanium	3.54	....	3260±	....	.....	....	13.7	......
Tungsten	18.8	.033	6152	....	.....	....	23	500,000
Zinc	7.14	.096	786.9	1724	.0088	28.1	26	9,000 – 24,000

From: Useful Data for Electrical Men - General Electric Co.

APPENDIX C (Contd.)

TABLE 17

TEMPERATURE CONVERSIONS

The middle column of figures (in bold type) contains the reading (F or C) to be converted. If converting from degrees Fahrenheit to degrees Centigrade read the Centigrade equivalent in the column headed "C". If converting from degrees Centigrade to degrees Fahrenheit, read the Fahrenheit equivalent in the column headed "F".

F		C	F		C	F		C	F		C
-126.4	**-88**	-66.67	125.6	**52**	11.11	377.6	**192**	88.89	[illegible]	**466**	[illegible]
-122.8	**-86**	-65.56	129.2	**54**	12.22	381.2	**194**	90.00	874.4	**468**	[illegible]
-119.2	**-84**	-64.44	132.8	**56**	13.33	384.8	**196**	91.11	[illegible]	**470**	[illegible]
-115.6	**-82**	-63.33	136.4	**58**	14.44	388.4	**198**	92.22	[illegible]	**472**	244.44
-112.0	**-80**	-62.22	140.0	**60**	15.56	392.0	**200**	93.33	885.2	**474**	[illegible]
-108.4	**-78**	-61.11	143.6	**62**	16.67	395.6	**202**	94.44	[illegible]	**476**	[illegible]
-104.8	**-76**	-60.00	147.2	**64**	17.78	399.2	**204**	95.56	[illegible]	**478**	[illegible]
-101.2	**-74**	-58.89	150.8	**66**	18.89	402.8	**206**	96.67	[illegible]	**480**	[illegible]
-97.6	**-72**	-57.78	154.4	**68**	20.00	406.4	**208**	97.78	[illegible]	**482**	[illegible]
-94.0	**-70**	-56.67	158.0	**70**	21.11	410.0	**210**	98.89	903.2	**484**	251.11
-90.4	**-68**	-55.56	161.6	**72**	22.22	413.6	**212**	100.00	906.8	**486**	252.22
-86.8	**-66**	-54.44	165.2	**74**	23.33	417.2	**214**	101.11	910.4	**488**	[illegible]
-83.2	**-64**	-53.33	168.8	**76**	24.44	420.8	**216**	102.22	914.0	**490**	254.44
-79.6	**-62**	-52.22	172.4	**78**	25.56	424.4	**218**	103.33	917.6	**492**	[illegible]
-76.0	**-60**	-51.11	176.0	**80**	26.67	428.0	**220**	104.44	921.2	**494**	256.67
-72.4	**-58**	-50.00	179.6	**82**	27.78	431.6	**222**	105.56	[illegible]	**496**	257.78
-68.8	**-56**	-48.89	183.2	**84**	28.89	435.2	**224**	106.67	[illegible]	**498**	258.89
-65.2	**-54**	-47.78	186.8	**86**	30.00	438.8	**226**	107.78	[illegible]	**500**	260.00
-61.6	**-52**	-46.67	190.4	**88**	31.11	442.4	**228**	108.89	[illegible]	**502**	261.11
-58.0	**-50**	-45.56	194.0	**90**	32.22	446.0	**230**	110.00	939.2	**504**	262.22
-54.4	**-48**	-44.44	197.6	**92**	33.33	449.6	**232**	111.11	942.8	**506**	263.33
-50.8	**-46**	-43.33	201.2	**94**	34.44	453.2	**234**	112.22	946.4	**508**	264.44
-47.2	**-44**	-42.22	204.8	**96**	35.56	456.8	**236**	113.33	[illegible]	**510**	[illegible]
-43.6	**-42**	-41.11	208.4	**98**	36.67	460.4	**238**	114.44	[illegible]	**512**	[illegible]
-40.0	**-40**	-40.00	212.0	**100**	37.78	464.0	**240**	115.56	957.2	**514**	267.78
-36.4	**-38**	-38.89	215.6	**102**	38.89	467.6	**242**	116.67	960.8	**516**	268.89
-32.8	**-36**	-37.78	219.2	**104**	40.00	471.2	**244**	117.78	964.4	**518**	[illegible]
-29.2	**-34**	-36.67	222.8	**106**	41.11	474.8	**246**	118.89	968.0	**520**	271.11
-25.6	**-32**	-35.56	226.4	**108**	42.22	478.4	**248**	120.00	[illegible]	**522**	[illegible]
-22.0	**-30**	-34.44	230.0	**110**	43.33	482.0	**250**	121.11	975.2	**524**	273.33
-18.4	**-28**	-33.33	233.6	**112**	44.44	485.6	**252**	122.22	978.8	**526**	274.44
-14.8	**-26**	-32.22	237.2	**114**	45.56	489.2	**254**	123.33	[illegible]	**528**	[illegible]
-11.2	**-24**	-31.11	240.8	**116**	46.67	492.8	**256**	124.44	[illegible]	**530**	276.67
-7.6	**-22**	-30.00	244.4	**118**	47.78	496.4	**258**	125.56	[illegible]	**532**	[illegible]
-4.0	**-20**	-28.89	248.0	**120**	48.89	500.0	**260**	126.67	993.2	**534**	[illegible]
-0.4	**-18**	-27.78	251.6	**122**	50.00	503.6	**262**	127.78	996.8	**536**	280.00
+3.2	**-16**	-26.67	255.2	**124**	51.11	507.2	**264**	128.89	1000.4	**538**	281.11
+6.8	**-14**	-25.56	258.8	**126**	52.22	510.8	**266**	130.00	1004.0	**540**	282.22
+10.4	**-12**	-24.44	262.4	**128**	53.33	514.4	**268**	131.11	1007.6	**542**	283.33
+14.0	**-10**	-23.33	266.0	**130**	54.44	518.0	**270**	132.22	1011.2	**544**	284.44
+17.6	**-8**	-22.22	269.6	**132**	55.56	521.6	**272**	133.33	1014.8	**546**	[illegible]
+21.2	**-6**	-21.11	273.2	**134**	56.67	525.2	**274**	134.44	1018.4	**548**	[illegible]
+24.8	**-4**	-20.00	276.8	**136**	57.78	528.8	**276**	135.56	1022.0	**550**	287.78
+28.4	**-2**	-18.89	280.4	**138**	58.89	532.4	**278**	136.67	[illegible]	**560**	[illegible]
+32.0	**±0**	-17.78	284.0	**140**	60.00	536.0	**280**	137.78	[illegible]	**570**	[illegible]
+35.6	**+2**	-16.67	287.6	**142**	61.11	539.6	**282**	138.89	1076.0	**580**	304.44
+39.2	**+4**	-15.56	291.2	**144**	62.22	543.2	**284**	140.00	[illegible]	**590**	[illegible]
+42.8	**+6**	-14.44	294.8	**146**	63.33	546.8	**286**	141.11	[illegible]	**600**	[illegible]
+46.4	**+8**	-13.33	298.4	**148**	64.44	550.4	**288**	[illegible]	[illegible]	**610**	[illegible]
+50.0	**+10**	-12.22	302.0	**150**	65.56	554.0	**290**	[illegible]	1148.0	**620**	326.67
+53.6	**+12**	-11.11	305.6	**152**	66.67	[illegible]	**292**	[illegible]	1166.0	**630**	[illegible]
+57.2	**+14**	-10.00	309.2	**154**	67.78	[illegible]	**294**	[illegible]	1184.0	**640**	[illegible]
+60.8	**+16**	-8.89	312.8	**156**	68.89	[illegible]	**296**	[illegible]	1202.0	**650**	[illegible]
+64.4	**+18**	-7.78	316.4	**158**	70.00	[illegible]	**298**	[illegible]	[illegible]	**660**	[illegible]
+68.0	**+20**	-6.67	320.0	**160**	71.11	[illegible]	**300**	148.89	[illegible]	**670**	[illegible]
+71.6	**+22**	-5.56	323.6	**162**	72.22	[illegible]	**302**	150.00	[illegible]	**680**	[illegible]
+75.2	**+24**	-4.44	327.2	**164**	73.33	[illegible]	**304**	151.11	[illegible]	**690**	[illegible]
+78.8	**+26**	-3.33	330.8	**166**	74.44	[illegible]	**306**	152.22	[illegible]	**700**	[illegible]
+82.4	**+28**	-2.22	334.4	**168**	75.56	[illegible]	**308**	[illegible]	[illegible]	**710**	[illegible]
+86.0	**+30**	-1.11	338.0	**170**	76.67	[illegible]	**310**	154.44	[illegible]	**720**	[illegible]
+89.6	**+32**	±0.00	341.6	**172**	77.78	[illegible]	**312**	155.56	[illegible]	**730**	387.78
+93.2	**+34**	+1.11	345.2	**174**	78.89	[illegible]	**314**	156.67	[illegible]	**740**	[illegible]
+96.8	**+36**	+2.22	348.8	**176**	80.00	[illegible]	**316**	[illegible]	[illegible]	**750**	[illegible]
+100.4	**+38**	+3.33	352.4	**178**	81.11	[illegible]	**318**	158.89	1400.0	**760**	[illegible]
+104.0	**+40**	+4.44	356.0	**180**	82.22	[illegible]	**320**	160.00	1418.0	**770**	410.00
107.6	**42**	5.56	359.6	**182**	83.33	[illegible]	**322**	161.11	[illegible]	**780**	[illegible]
111.2	**44**	6.67	363.2	**184**	84.44	[illegible]	**324**	162.22	[illegible]	**790**	[illegible]
114.8	**46**	7.78	366.8	**186**	85.56	[illegible]	**326**	163.33	[illegible]	**800**	[illegible]
118.4	**48**	8.89	370.4	**188**	86.67	622.4	**328**	164.44	[illegible]	**810**	[illegible]
122.0	**50**	10.00	374.0	**190**	87.78	626.0	**330**	165.56	[illegible]	**820**	[illegible]

APPENDIX C (Contd.)

TABLE 18

Properties of Air*

Temperature °F	Temperature °C	c ** Specific Heat, Btu/(lb.)(°F)	*** ρ Density, lb./cu.ft.	μ, Viscosity, lb./(ft.)(hr.)	k, Thermal Conductivity, Btu/(hr.)(ft.)(°F)	$\frac{c\mu}{k}$ Prandtl Number	β Coeff. of Thermal Expansion $\frac{1}{°R}$	*** a × 10[illegible] Free Convection Modulus $\frac{1}{(cu.ft.)(°F)}$
-40	-40	0.239	0.0968	[illegible]	0.0116	0.74	0.00244	[illegible]
0	-17.8	0.239	0.0863	0.040	0.0132	0.72	0.00217	3.00
50	[illegible]	0.240	0.0779	0.043	0.0145	0.71	0.00196	[illegible]
[illegible]	[illegible]	0.240	0.0708	0.046	0.0158	0.70	0.00179	[illegible]
[illegible]	[illegible]	0.241	0.0651	0.049	0.0170	0.70	[illegible]	[illegible]
[illegible]	93.3	0.241	0.0601	0.052	0.0182	0.69	[illegible]	[illegible]
250	121.1	0.242	0.0559	0.055	0.0192	0.68	[illegible]	[illegible]
300	148.9	0.242	0.0522	0.058	0.0240	0.68	[illegible]	[illegible]
350	176.7	0.243	0.0490	0.060	0.0216	[illegible]	0.00123	[illegible]
400	204.4	0.245	0.0461	0.062	0.0227	0.67	0.00116	[illegible]
450	232.2	0.246	0.0436	0.065	0.0239	0.67	0.00110	[illegible]
500	260.0	0.247	0.0413	0.067	0.0250	0.66	0.00104	[illegible]
550	288.0	0.249	0.0393	0.070	0.0264	0.66	0.00099	[illegible]
600	315.0	0.250	0.0374	0.072	0.0271	[illegible]	[illegible]	[illegible]
650	343.0	0.252	0.0358	0.074	0.0282	[illegible]	0.00090	[illegible]
700	371.1	[illegible]	0.0342	[illegible]	0.0291	[illegible]	[illegible]	[illegible]

* Table derived mainly from Ref. 11

** Specific Heat at Constant Pressure

*** Density and Convection Modulus for Atmospheric Pressure (29.92 in. Hg.)

APPENDIX C (Contd.)

TABLE 19

Properties of Air, Helium, Hydrogen, and Nitrogen at 100°C. (212°F.)

	c $\frac{\text{Btu}}{\text{(lb.)(°F)}}$	ρ $\frac{\text{lb.}}{\text{cu.ft.}}$	μ $\frac{\text{lb.}}{\text{(ft.)(hr.)}}$	k $\frac{\text{Btu}}{\text{(hr.)(ft.)(°F)}}$	$\frac{c\mu}{k}$	a $\frac{1}{\text{(cu.ft.)(°F)}}$
Air	0.241	0.0591	0.0527	0.0184	0.689	539,000
Helium	1.25	0.00816	0.0544	0.097	0.700	9,810
Hydrogen	3.43	0.00411	0.0254	0.129	0.676	11,000
Nitrogen	0.25	0.0571	0.0507	0.0180	0.704	566,000

From: Several sources

APPENDIX C (Contd.)

TABLE 20

Emissivity Values of Common Materials

Materials	*Emissivity at Low Temperature
Lampblack	0.95
Dull-oxide-type paints	0.94
Asbestos and most nonmetallic insulating materials	0.93
Most glass-type paint & enamels	0.88
Oxidized steel	0.75
Oxidized copper	0.70
Oxidized brass	0.60
Aluminum paint	0.27 to 0.67
Oxidized nickel or monel	0.42
Anodized aluminum) (Oxidized aluminum)	0.22 to 0.40 normally, but may vary from 0.05 to 0.75, depending on thickness of film.

*For higher temperature values, see table 21.

From: Ref. 4

TABLE 21

Total Emissivity Values for Various Metals & Glasses*

Material	Condition	At 100°C	At 320°C	At 500°C
Alleghany metal	No. 4 Polish	.13		
Alleghany alloy No. 66	Polished	.11		
Aluminum	Commercial Sheet	.09		
"	Polished	.095		
"	Rough Polish	.18		
Brass	Polished	.059		
Carbon	Rough Plate	.77	.77	.72
Carbon, graphitized	" "	.76	.75	[illegible]
Chromium	Polished	.075		
Copper	Polished	.052 to .04		
Copper - nickel	Polished	.059		
Iron	Dark Gray Surface	.31		
Iron	Roughly Polished	.27		
Lampblack	Rough Deposit	.84		.78
Molybdenum	Polished	.071		
Nickel	Polished	.072		
Nickel-silver	Polished	.135		
Radiator Paint, white	Clean	.79		
" " , cream	"	.77		
" " , black	"	.84		
" " , bronze	"	.51		
Silver	Polished	.052 to .03		
Stainless steel	Polished	.074		
Steel	Polished	.066		
Tin	Polished	.069		
Tin	Commercial Coat	.084		
Tungsten	Polished Coat	.066		
Zinc	Commercial Coat	.21		
Fuzed quartz	1.96 mm. thick	.775	.76	.67
Covex D (glass)	3.40 mm. thick	.83	.90	.91
Nonex (glass)	1.57 mm. thick	.835	.87	.82
Aluminum paint		.29		

* (From Ref. 12)

APPENDIX C (Contd.)

TABLE 22

Properties of Water

Temp. °F	Temp. °C	c Btu/(lb.)(°F)	μ lb./(ft.)(hr.)	k Btu/(hr.)(ft.)(°F)	cu/k	a x 10^{-8} 1/(cu.ft.)(°F)
32	0.0	1.009	4.33	0.327	13.4	
40	4.4	1.005	3.75	0.332	11.3	0.3
50	10.0	1.002	3.17	0.338	9.4	1.0
60	15.6	1.000	2.71	0.344	7.9	1.7
70	21.1	0.998	2.37	0.349	6.8	2.3
80	26.7	0.998	2.08	0.355	5.8	3.0
90	32.2	0.997	1.85	0.360	5.1	3.9
100	37.8	0.997	1.65	0.364	4.5	5.2
110	43.3	0.997	1.49	0.368	4.0	6.6
120	48.9	0.997	1.36	0.372	3.6	7.7
130	54.4	0.998	1.24	0.375	3.3	8.9
140	60.0	0.998	1.14	0.378	3.0	10.2
150	65.6	0.999	1.04	0.381	2.7	12.0
160	71.1	1.000	0.97	0.384	2.5	13.9
170	76.7	1.001	0.90	0.386	2.3	15.5
180	82.2	1.002	0.84	0.389	2.2	17.1
190	87.8	1.003	0.79	0.390	2.1	
200	93.3	1.004	0.74	0.392	1.9	

From: Ref. 11

APPENDIX C (Contd.)

TABLE 23

Properties of Silicone Fluid -(DC550-112 Centistoke Grade)

Temp. °F	°C	c Btu / (lb.)(°F)	μ lb. / (ft.)(hr.)	k Btu / (hr.)(ft.)(°F)	lb. / cu.ft.	a x 10^{-8} 1 / (cu.ft.)(°F)
60	15.6	0.378		0.0783	67.0	
80	26.7	0.386		0.0778	66.4	
100	32.2	0.395		0.0773	65.9	
120	37.8	0.405	132	0.0767	64.3	0.32
140	60.0	0.414	98.0	0.0761	64.8	0.43
160	71.1	0.425	74.0	0.0755	64.2	0.59
180	82.2	0.437	57.0	0.0749	63.7	0.78
200	93.3	0.452	44.0	0.0742	63.2	0.99
220	104.4	0.470	36.0	0.0736	62.7	1.22
240	115.6	0.487	30.0	0.0729	62.2	1.45
260	126.7	0.501	25.5	0.0721	61.7	1.69
280	137.8	0.514	22.1	0.0714	61.2	1.94
300	148.9	0.523	19.6	0.0707	60.8	2.19
320	160.0	0.531	17.3	0.0700	60.3	2.45
340	171.1	0.538	15.9	0.0692	59.8	2.72
360	182.2	0.544	14.3	0.0685	59.4	2.99

Flash point min.	300°C
Freezing point	-50°C
Coefficient of expansion, K x 1000/°C (25 to 100°C)	.75
Dielectric strength	800 volts/mil
Power factor at 25°C	
10^2 cps	.00158
10^6 cps	.00003
Dielectric constant at 25°C at 10^2 and 10^6 cps	2.9
Color	Slightly yellow

Courtesy, Dow Corning Corp.

APPENDIX C (Contd.)

TABLE 24

Properties of DC-701 Silicone Fluid

Color	Clear water white
Viscosity at 25°C	10 centistokes
Freezing point	-60°C
Boiling point at atmospheric pressure	340°C
Flash point, min.	143°C
Specific gravity at 25°C	1.04
Coefficient of expansion per 1000/°C, 0°-100°C	.93
Dielectric strength, volts per mil	500
Dielectric constant at 25°C and 10^2 and 10^5 cps	2.63
Power factor at 25°C	
at 10^2 cps	.0002
at 10^5 cps	.0001

Courtesy - Dow Corning Corp.

TABLE 25

PHYSICAL PROPERTIES OF DOW CORNING 200 FLUIDS

Viscosity in Cstks. at 25°C	Viscosity in SSU at 100°F	Viscosity Temperature Coefficient	Die-lectric Constant	Freezing Point °C	Freezing Point °F	Boiling Point Temperature °C	Boiling Point Temperature °F	Press mm Hg	Flash Point Minimum °F	Thermal Conductivity	Specific Gravity 25°C/25°C
0.65	28	0.31	2.18	-68	-90	99.5	211	760	30	.00024	0.761
1.0	29	0.37	2.32	-86	-123	152	305	760	110	.00024	0.818
1.5	30	0.46	2.40	-76	-105	192	377	760	160	.00025	0.853
2.0	31	0.48	2.45	-84	-119	230	446	760	175	.00026	0.873
				Pour Point							
3.0	33	0.51	2.52	-65	-85	70-100	158-212	0.5	215	.00027	0.900
5.0	39	0.55	2.58	-65	-85	120-160	248-320	0.5	275	.00028	0.920
10	52	0.57	2.65	-65	-85	> 200	> 392	0.5	325	.00032	0.940
20	80	0.59	2.68	-60	-76	> 200	> 392	0.5	520	.00034	0.955
50	185	0.59	2.72	-55	-67	> 250	> 482	0.5	535	.00036	0.960
						Volatility after 48 hrs. at °C	at °F	%			
100	350	0.60	2.74	-55	-67	200	392	< 2	575	.00037	0.970
200	720	0.62	2.74	-53	-63	200	392	< 2	600	.00037	0.971
350	1,250	0.62	2.75	-50	-58	200	392	< 2	600	.00038	0.972
500	1,750	0.62	2.75	-50	-58	200	392	< 2	600	.00038	0.972
1,000	3,500	0.62	2.76	-50	-58	200	392	< 2	600	.00038	0.973
				Solidification Temperature							
12,500	45,000	0.58	2.82	-46	-51	200	392	< 2	600	.00038	0.974
30,000	115,000	0.61	2.77	-44	-47	200	392	< 2	600		0.975

Courtesy, Dow Corning Corp.

APPENDIX C (Contd.)

TABLE 26

Properties of Mineral Oil (10-C Transformer Oil)

Temp. °F	Temp. °C	c Btu / (lb.)(°F)	μ lb. / (ft.)(hr.)	k Btu / (hr.)(ft.)(°F)	a x 10^{-8} 1 / (cu.ft.)(°F)
60	15.6	41.7		0.0785	1.00
80	26.7	42.7	19.9		1.30
100	32.2	43.7	16.8		1.82
120	37.8	44.7	14.0		2.70
140	60.0	45.7	11.6		3.60
160	71.1	46.7	9.7		4.55
180	82.2	47.7	8.2		5.50
200	93.3	48.7	7.1		6.40
220	104.4	49.7	6.2		7.35
240	115.6	50.6	5.3		8.30
260	126.7	51.6	4.6		9.25
280	137.8	52.5	4.0		10.20
300	148.9	53.5	3.5		11.10
320	160.0	54.5	3.0		12.05
340	171.1	55.5	2.7		13.00
360	182.2	56.5	2.3		

Flash point	132°C
Fire point	149°C
Dielectric strength (new oil)	300 volts/mil
Dielectric constant at 1 M.C.	2.22

From: Pender, H. & Del Mar, W. - Electrical Engineers' Handbook, 4th Edition, 1951

APPENDIX C (Contd.)

TABLE 27

Typical Properties of Perfluorocarbon Liquids

Identification Number	FCX-326	FCX-327	FCX-328
Empirical Formula	C_7F_{14}	C_8F_{16}	$C_8F_{15}Cl$
Molecular Weight	350	400	416.5
Boiling Point (^{o}C.)	76	102	129
M.P.	Glass	Glass	Glass
Index of Refraction	1.2762 (30^{o})	1.2858 (30^{o})	11.3170 (25^{o})
Density	1.7999 (20^{o})	1.853 (20^{o})	1.869 (20^{o})
Dielectric Constant*	1.69-1.70	1.75	2.03
Power Factor*	0.0045-0.0005	0.0098-0.0010	0.0110-0.0008
Dielectric Strength (2) (Volts)	16,600	15,000	12,900
Direct Current Resistance (ohms/cm.^{3})	5.2×10^{12}	1.2×10^{12}	1.6×10^{12}

The following are the characteristics of Standard Transformer Oil:

Dielectric constant = 2.00 Power factor = 0.0053 - 0.0004
Resistivity = 1.1×10^{12} Dielectric strength = 14,400

*From 100 cycles to 100 kc.

Notes: (from ref. 24) Method ASTM D 117-43, modified using 0.054" gap.

Courtesy, duPont

APPENDIX C (Contd.)

TABLE 28

Typical Solubility Properties of Perfluorocarbon Liquids

	FCX-326 (C_7F_{14})	FCX-327 (C_8F_{16})	FCX-328 ($C_8F_{15}Cl$)
Solubility (2) (wt. % at 27°C)			
$CHCl_3$	7	10	Misc.
CCl_4	Misc.	43(3)	Misc.
CH_3OH	Insol.	2.5	3.8
Ethyl acetate	15	13	Misc.
Acetone	10	9	42
Petroleum ether	Misc.	Misc.	Misc.
Ethyl ether	Misc.	Misc.	Misc.
Benzene	3	4.6	21
O-dichloro-benzene	Insol.	Insol.	Insol.

Notes: (from ref. 24) Solubilities of less than 1.0% are reported as "insoluble".
(At which point solution separated into two substantially equal phases.)

Courtesy, duPont

APPENDIX C (Contd.)

TABLE 29

Properties of Fluorochemical N-43

Formula	$(C_4F_9)_3N$
Physical state at room temperature	Colorless liquid
Odor	Odorless
Formula weight	671
Boiling point	177°C (351°F)
Freezing point (Glass Point)	-66°C (-87°F)
Pour point	-50°C (-58°F)
Density (g/cc at 25°C, 77°F)	1.872
Viscosity (Centistokes at 25°C, 77°F)	2.74
Refractive index (25°C, 77°F)	1.2910
Surface tension (dynes/cm at 25°C, 77°F)	16.1
Coefficient of volume expansion (per °C) (25-40°C, 77-104°F)	1.2×10^{-3}
(140-160°C, 284-320°F)	2.1×10^{-3}
Specific heat (cal/g/°C at 25-40°C, 77°-104°F)	0.27
Heat of vaporization (cal/mole at b.p.)	11,100
(cal/g, Btu/lb)	16.5
Vapor pressure (mm Hg at 25°C, 77°F)	0.3
Trouton ratio	24.6
Dielectric strength (ASTM-D 877)	40 KV
Dielectric constant (100 cps at 25°C, 77°F)	1.86
Power factor (100 cps at 25°C, 77°F)	<0.0005
Volume-resistivity (ohm-cm at 25°C, 77°F)	$10^{14} - 10^{16}$

Courtesy, Minnesota Mining & Mfg. Co.

APPENDIX C (Contd.)

TABLE 30

Properties of Fluorochemical 0-75

Formula	$C_8F_{16}O$
Physical state at room temperature	Colorless liquid
Odor	Odorless
Formula weight	416
Boiling point	101°C (214°F)
Freezing point (Glass point)	-113°C (-171°F)
Pour point	-100°C (-148°F)
Density (g/cc at 25°C, 77°F)	1.760
Viscosity (centistokes at 25°C, 77°F)	0.82
Refractive index (25°C, 77°F)	1.276
Surface tension (dynes/cm at 25°C, 77°F)	15.2
Coefficient of volume expansion (per °C) (25-40°C, 77-104°F)	1.6×10^{-3}
(40-80°C, 104-176°F)	2.0×10^{-3}
Specific heat (cal/g/°C at 25-40°C, 77-104°F)	0.26
Heat of vaporization (cal/mole at b.p.)	8,700
(cal/g)	20.9
Thermal conductivity, Liquid (BTU/hr./sq.ft /°F.ft.) (25°C, 77°F)	0.071
(60°C, 140°F)	0.054
Dielectric strength (ASTM-D877)	37 KV
Dielectric constant (100 cps at 25°C, 77°F)	1.85
Power factor (100 cps at 25°C, 77°F)	<0.0005
Volume resistivity (ohm-cm at 25°C, 77°F)	10^{14}-10^{16}

Courtesy, Minnesota Mining & Mfg. Co.

APPENDIX C (Contd.)

TABLE 31 COOLANT PROPERTIES	FREON 113 $C_2Cl_3F_3$	FREON 11 CCl_3F	TRICHLORETHYLENE C_2HCl_3	DICHLORETHYLENE $C_2H_2Cl_2$	METHYLENE CHLORIDE CH_2Cl_2	METHYL FORMATE HCO_2CH_3	ETHANOL C_2H_5OH	METHANOL CH_3OH	WATER H_2O	WATER-METHANOL MOL PER CENT 59.4 + 40.6	WATER-AMMONIA MOL PER CENT 71.7 + 28.3
Freezing Point °C	-35	-111	-72.8	-56.6	-96.6	-99.8	-117	-97.9	0	-65	-65
Boiling Point (Sea Level) °C	47.6	23.7	86.8	47.8	39.8	31.8	78.4	64.4	100	75	31.4
Average Density Lb. per cu. ft.	97.0	91.4	90.9	77.4	83.3	60.1	51.2	49.5	62.4	58.2	56.2
Latent Heat (Sea Level) Btu per Lb.	63	84	103	136	162	231	367	472	970	711	977
Vapor Pressure at 100°C Lb. per sq. in.	60.3	118	21.8	61.8	78	82.0	32.1	50.7	14.7	34.3 initial	124 initial
Latent heat (Sea Level) 1000 Btu per cu. ft.	6.1	7.7	9.46	10.5	13.4	13.9	18.8	23.4	60.5	41.4	55.7
Required Coolant Weight Lb. per kw-hr	53.4	40.6	32.2	25.1	21.0	14.7	9.31	7.24	3.52	4.8	3.49
Required Coolant Volume cu. ft. per kw-hr	0.56	0.444	0.36	0.325	0.254	0.246	0..82	0.146	0.0564	0.0825	0.0619
Underwriter's Classification	5	5			4	4	3				

From Reference 26

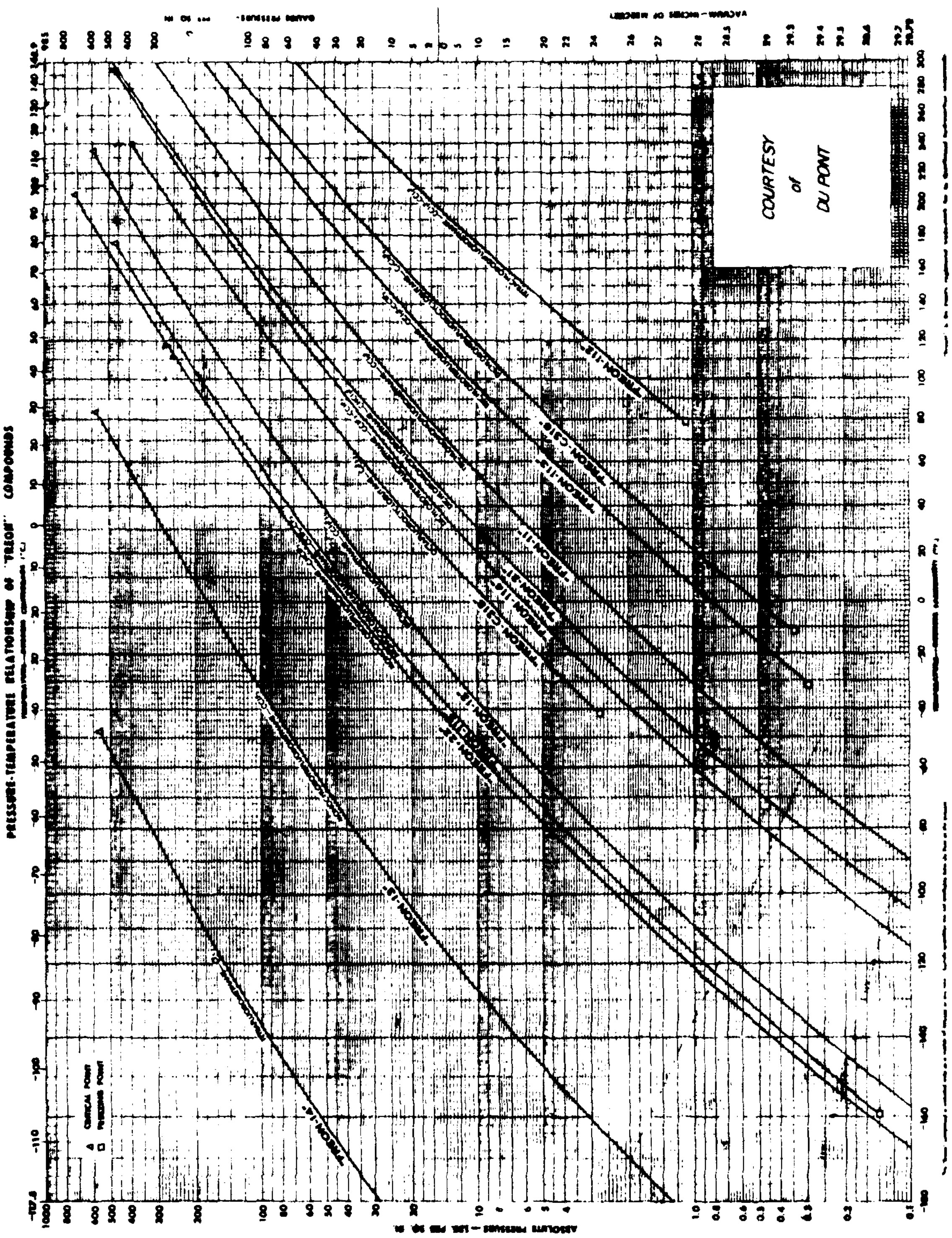

PRESSURE-TEMPERATURE RELATIONSHIP OF "FREON" COMPOUNDS
COURTESY
of
DU PONT

Fig. 56

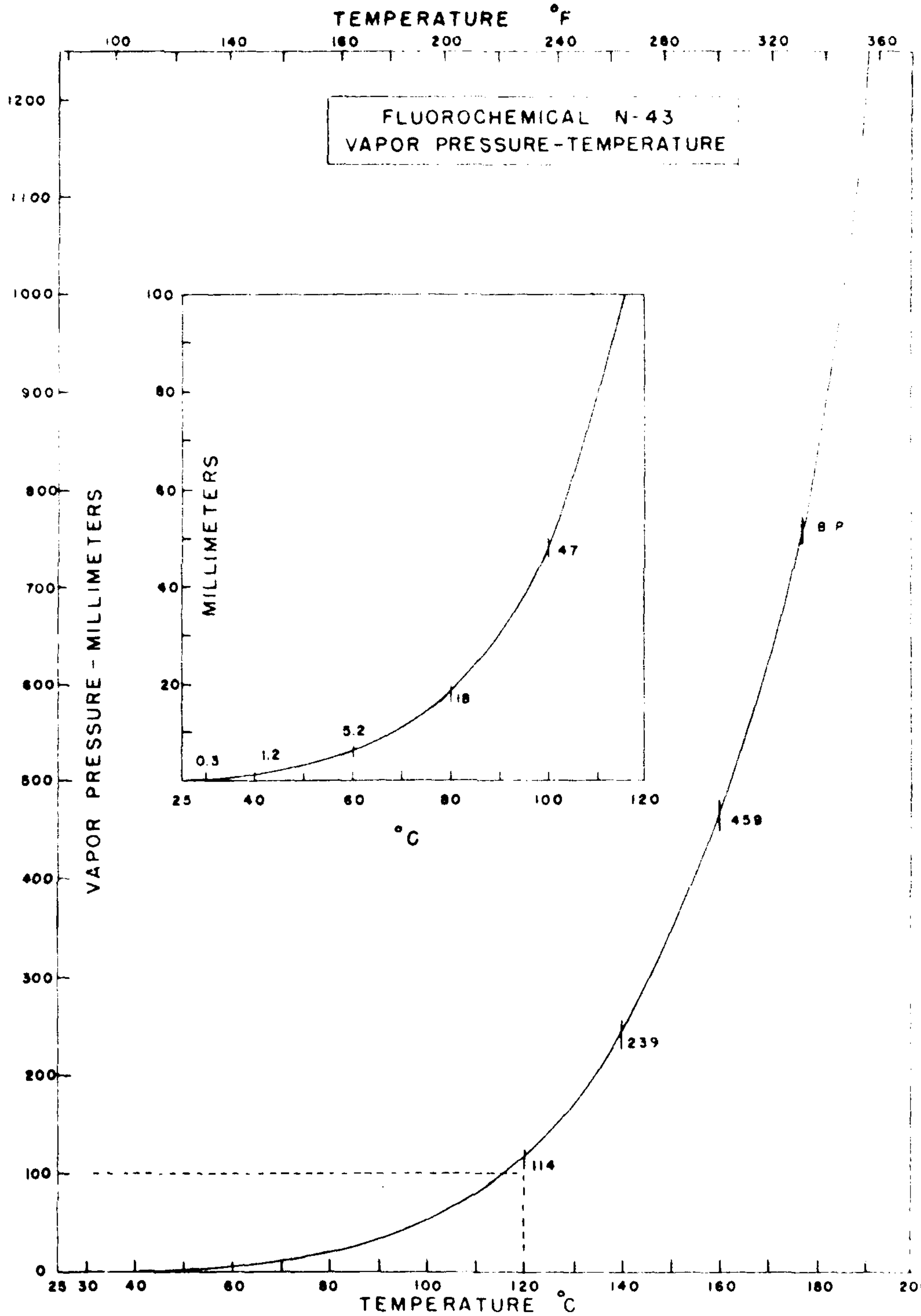
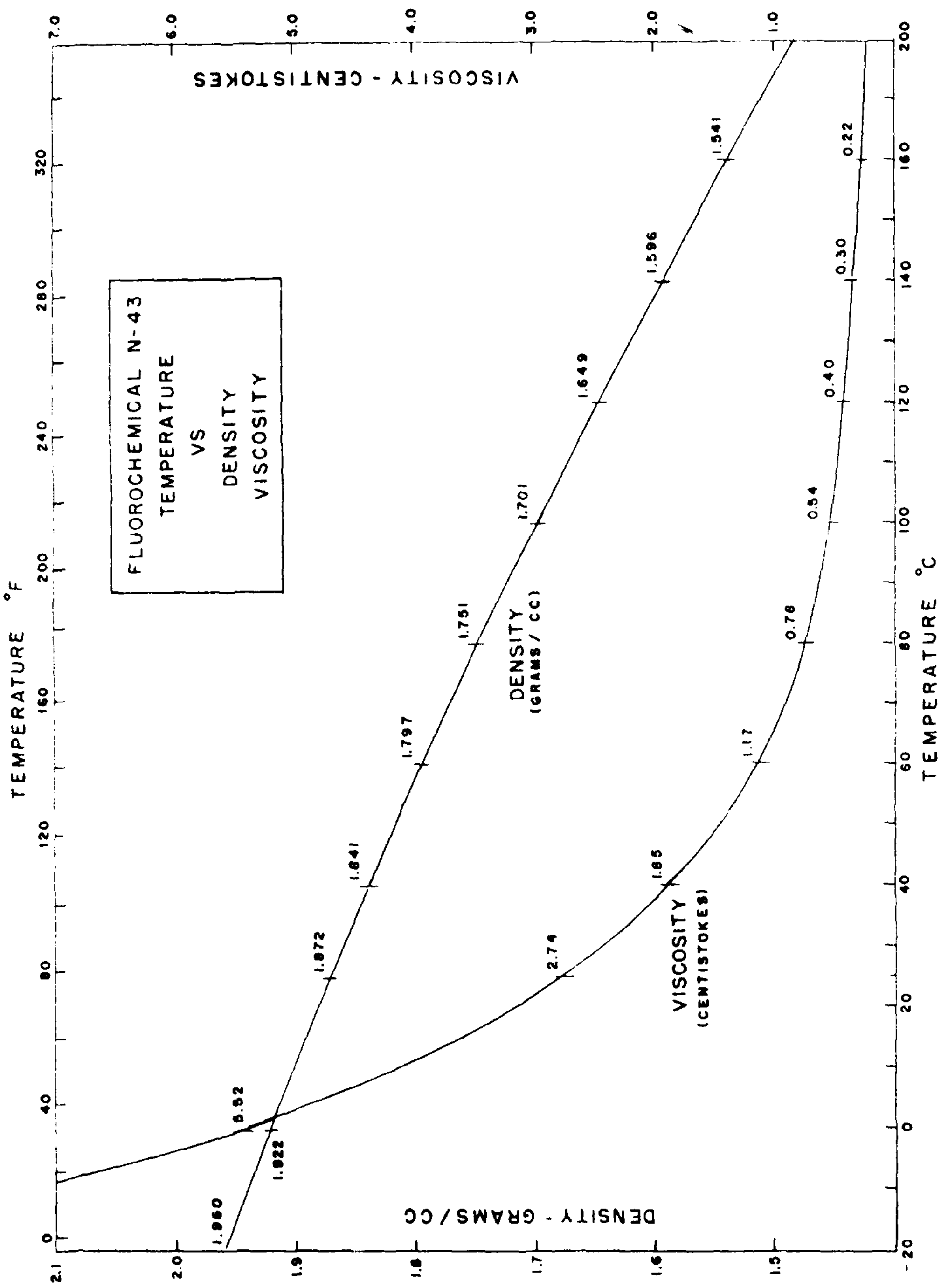


Fig. 57

Courtesy, Minnesota Mining & Mfg.



Courtesy, Minnesota Mining
and Mfg. Co.

Fig 58

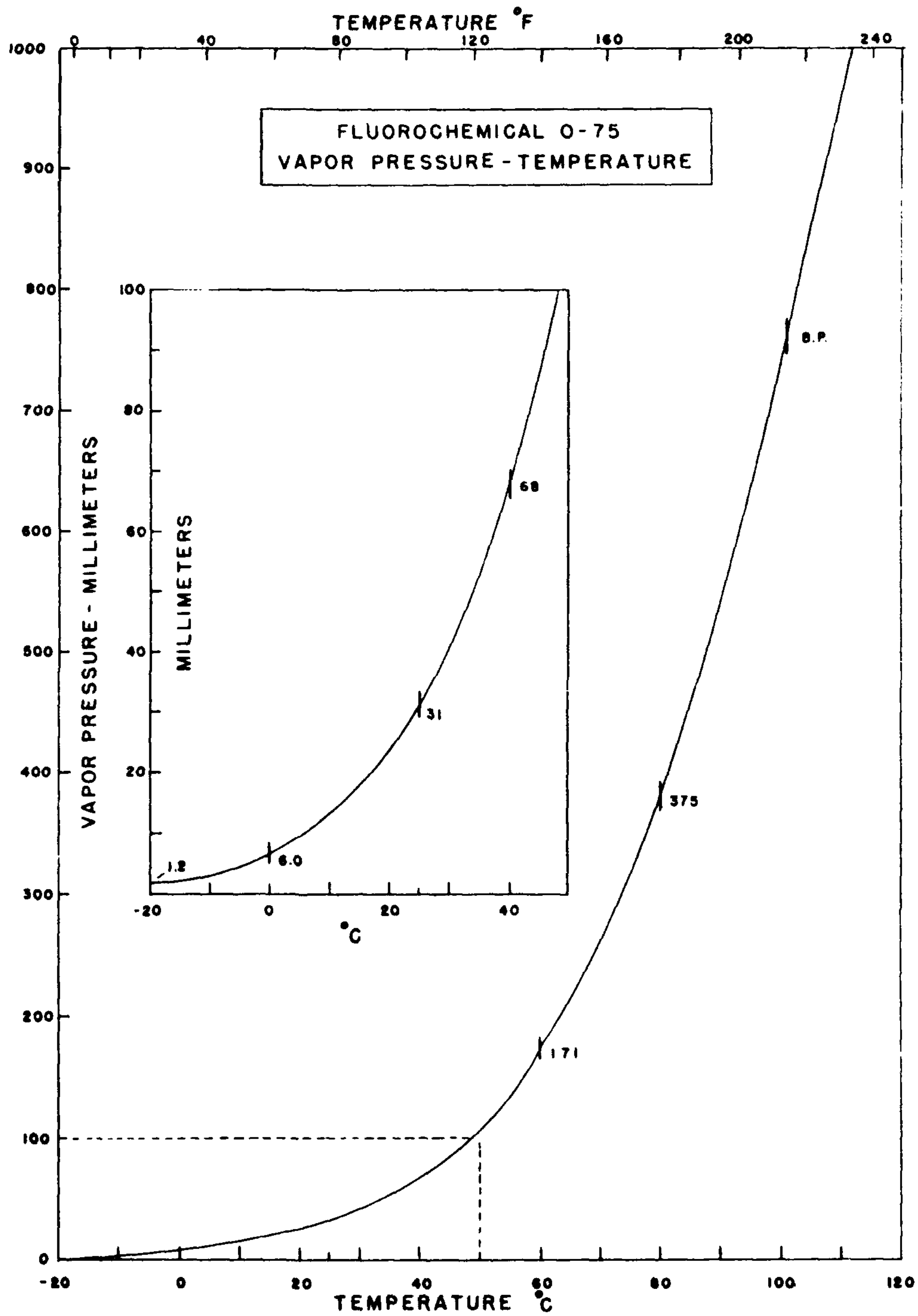


Fig. 59

Courtesy, Minnesota Mining & Mfg. Co.

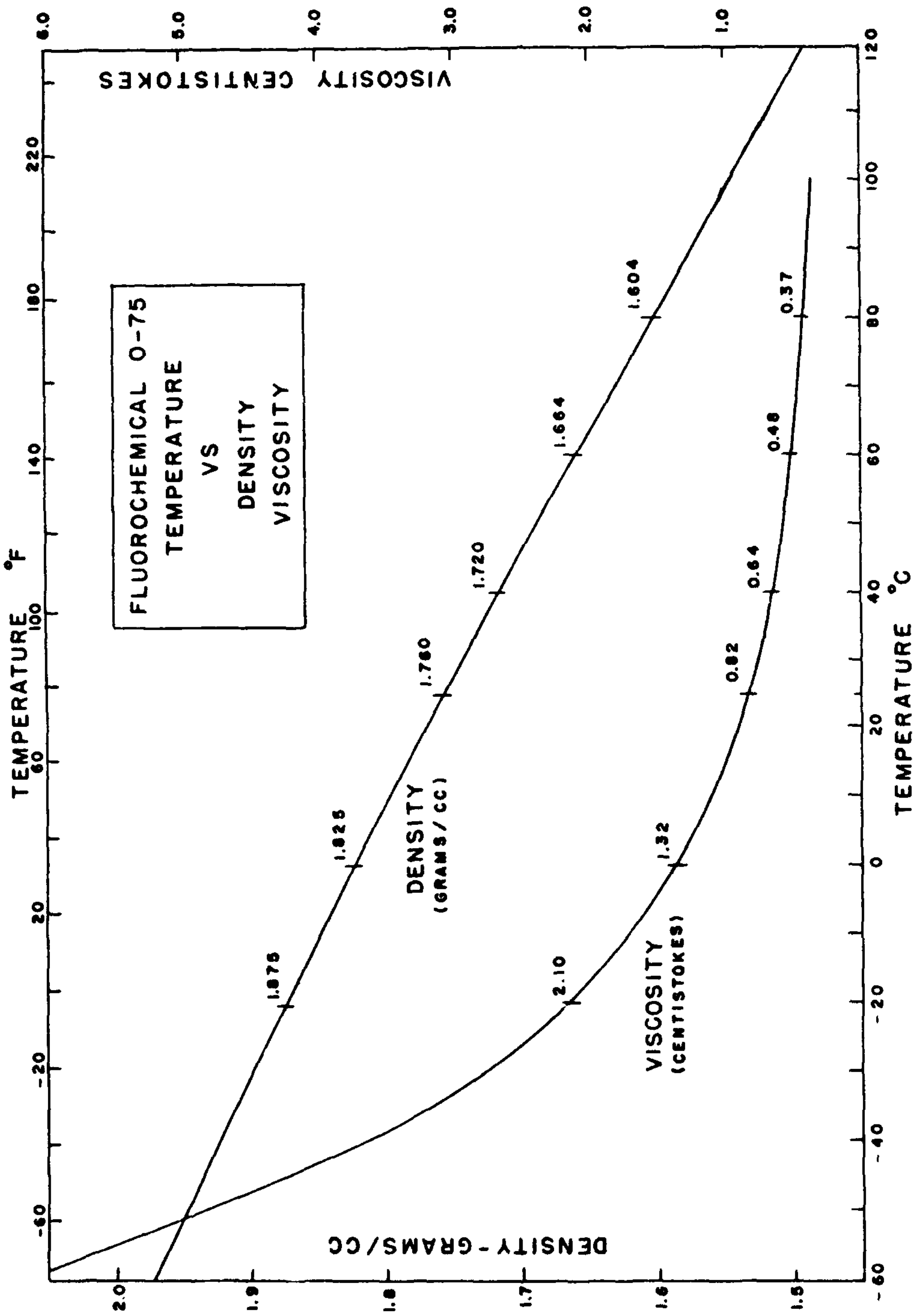


Fig. 60

Courtesy, Minnesota Mining & Mfg. Co.

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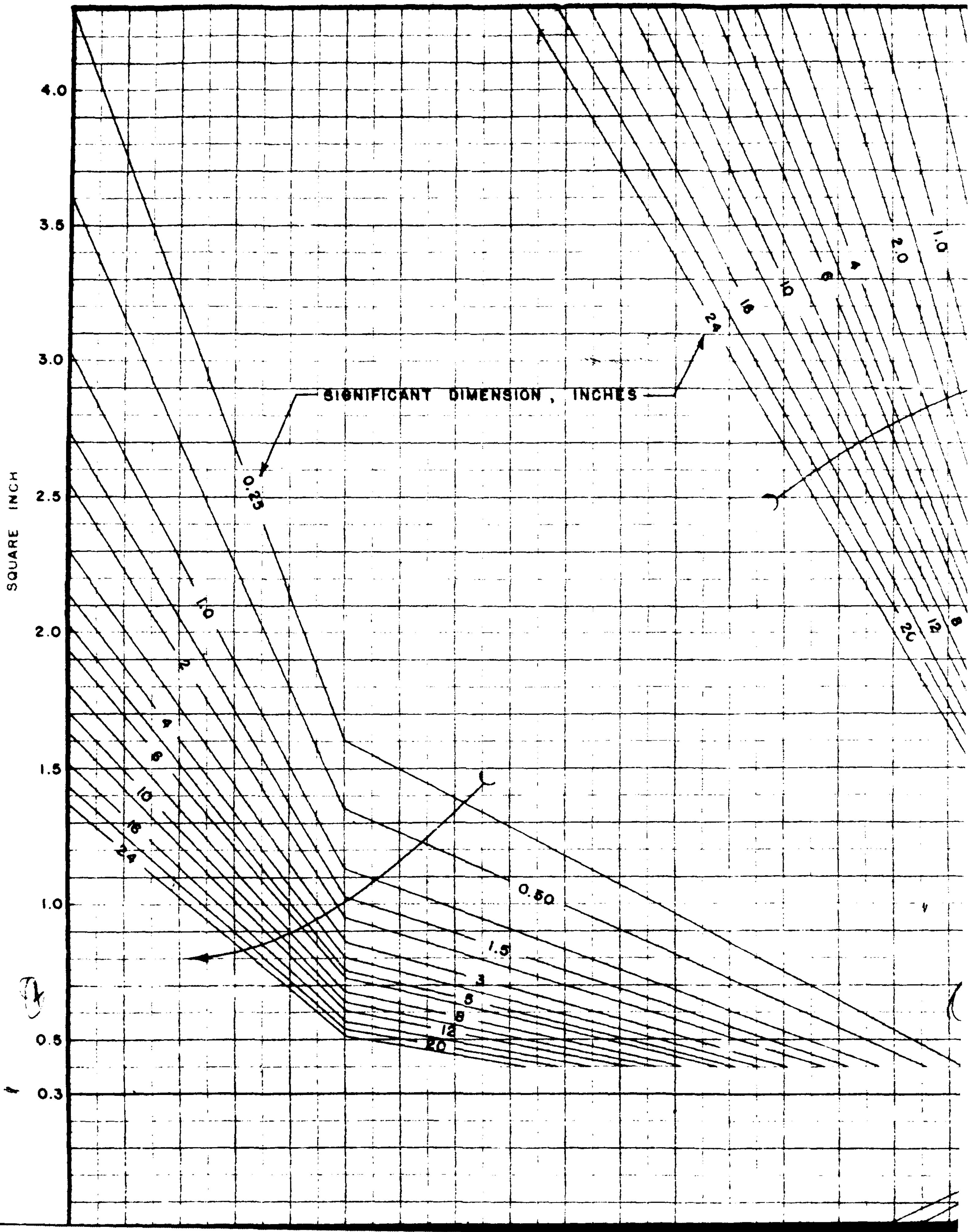

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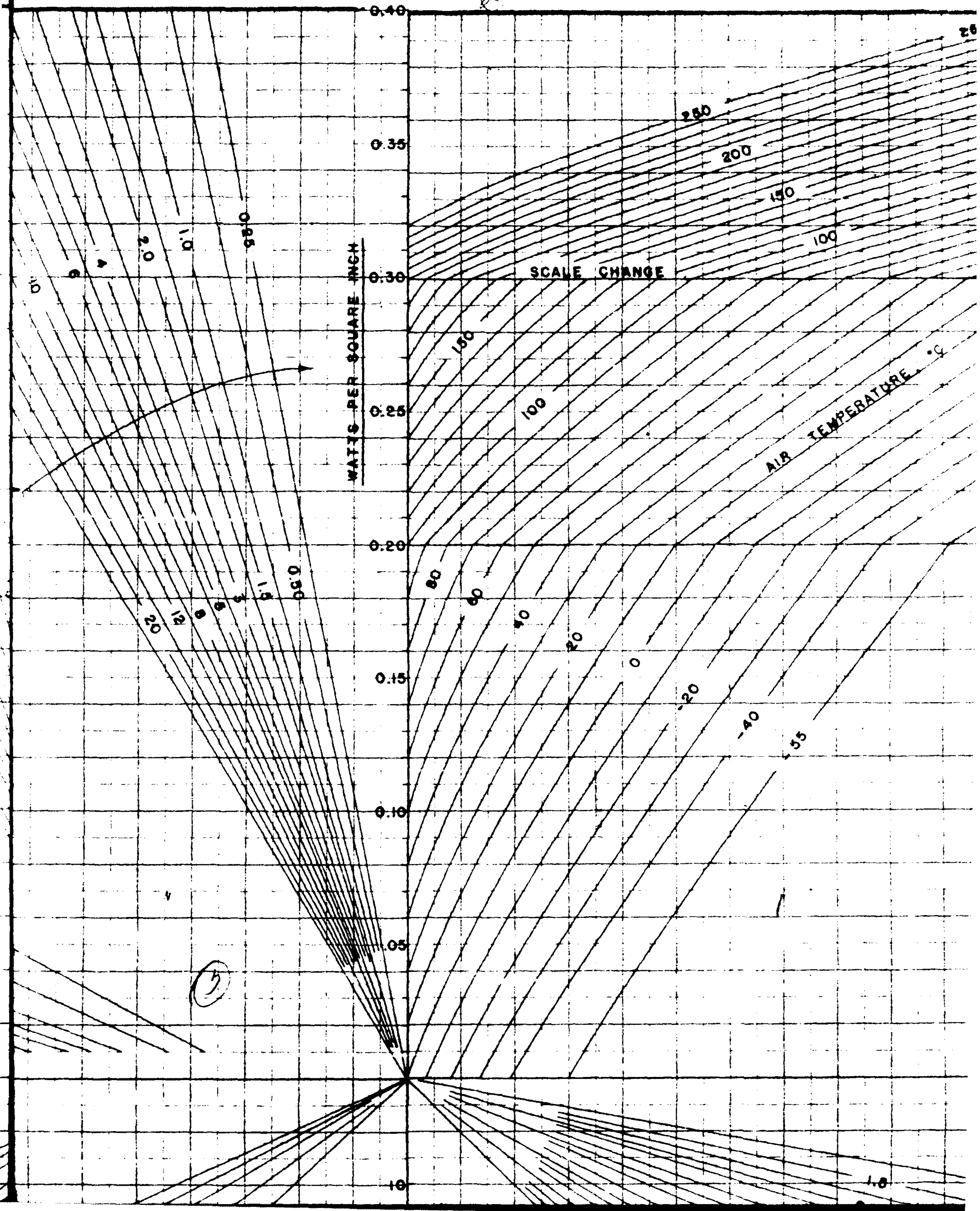

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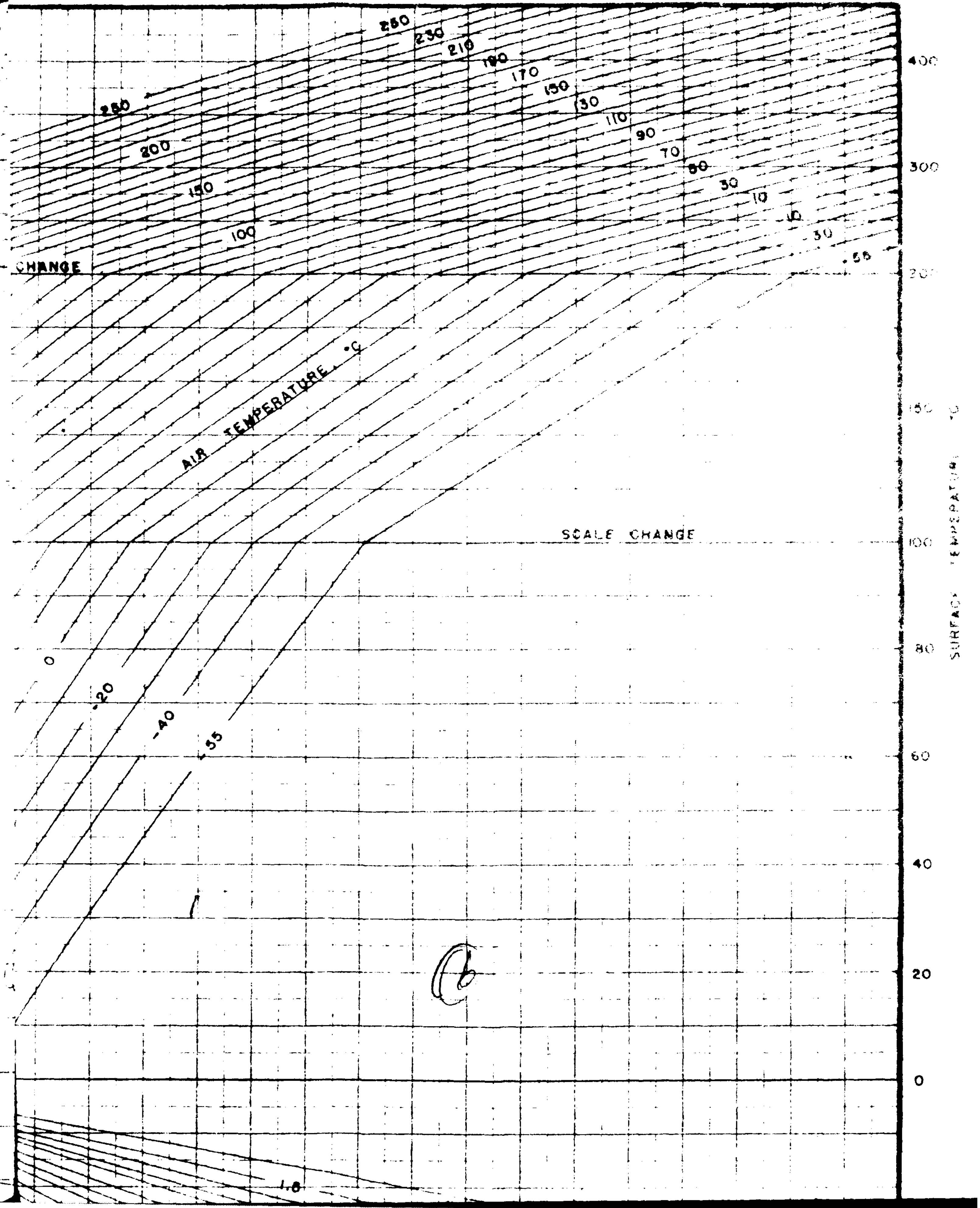
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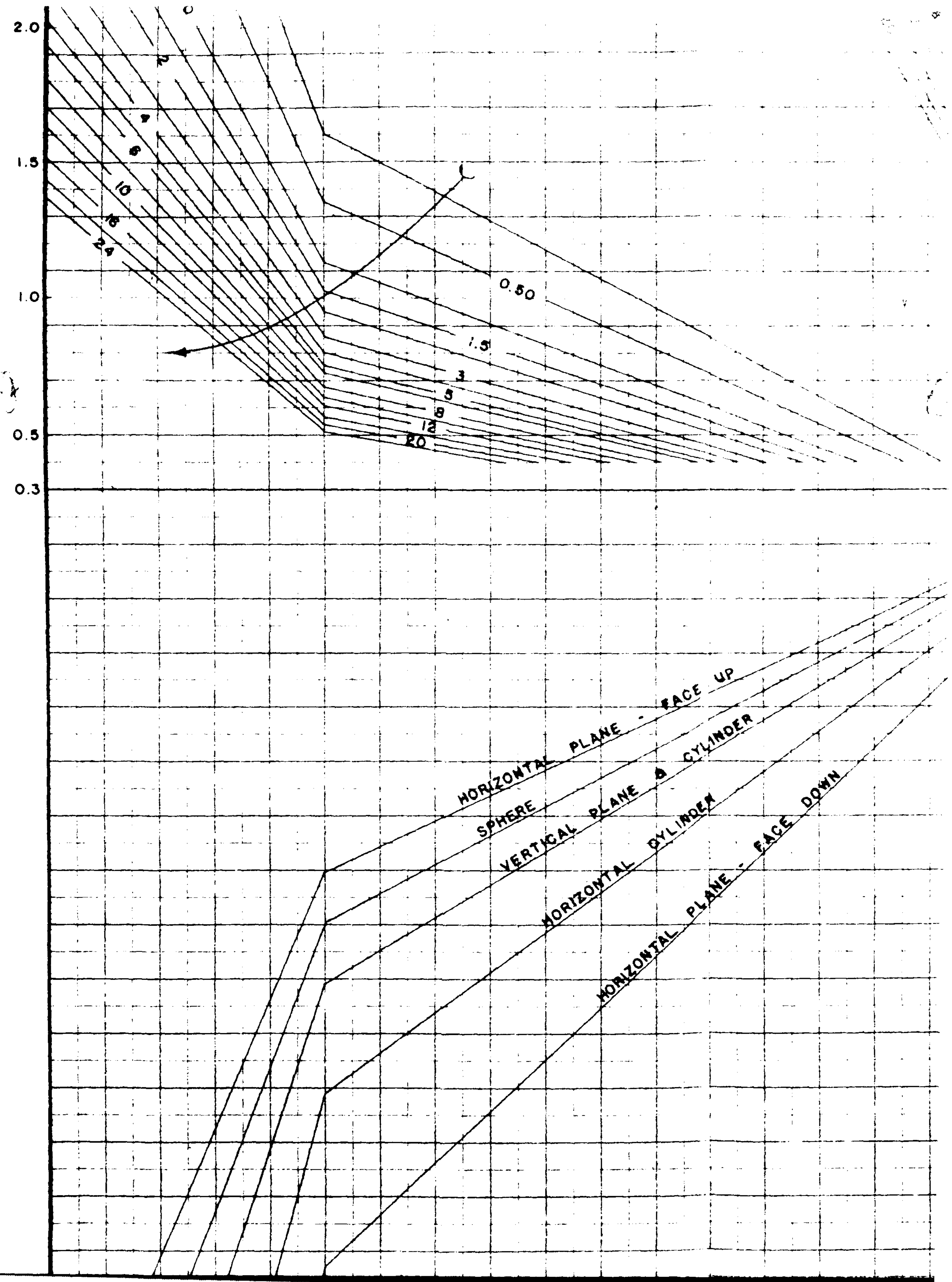
SURFACE TEMPERATURE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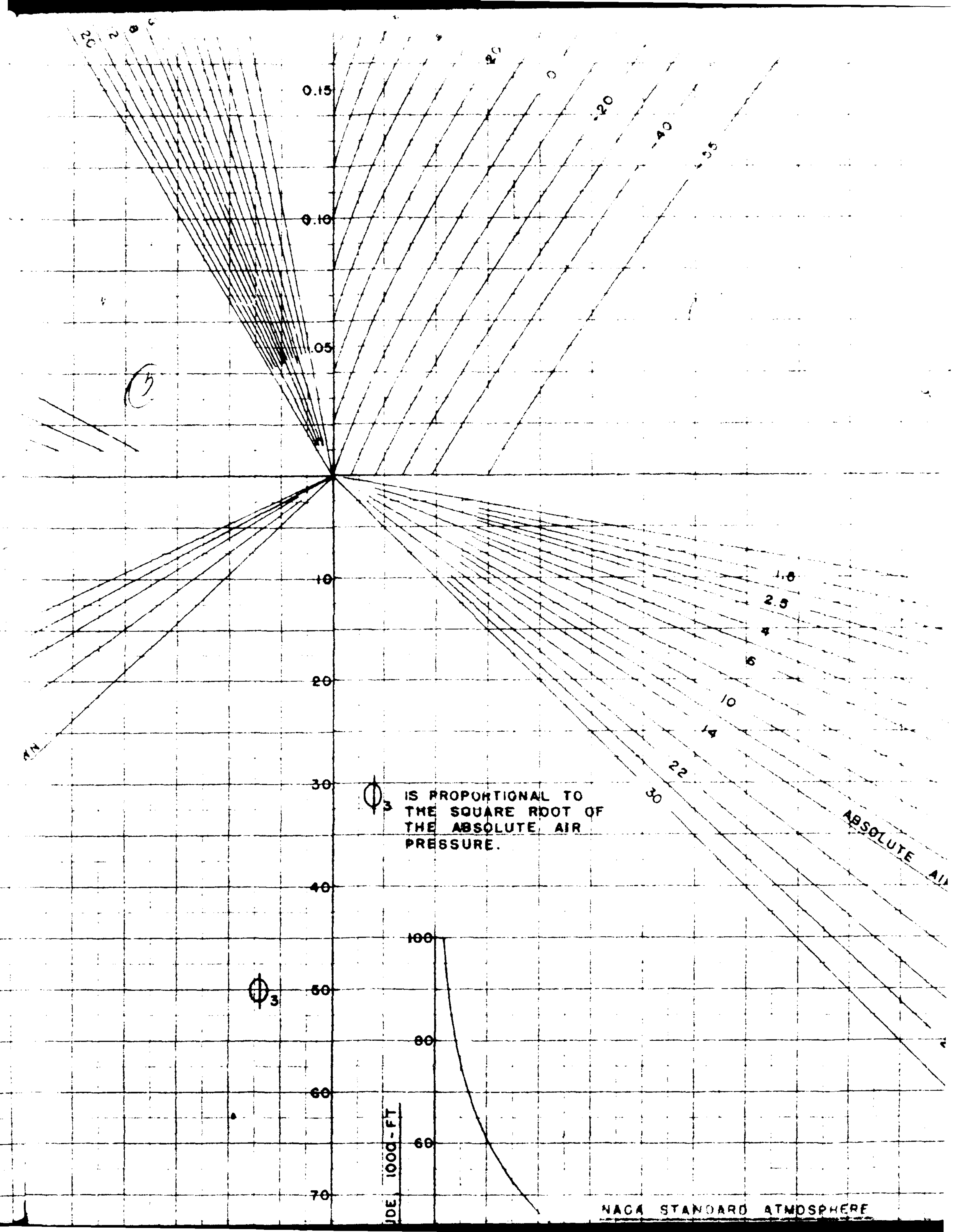
SPHERE
VERTICAL PLANE & CYLINDER
HORIZONTAL CYLINDER
HORIZONTAL PLANE - FACE DOWN



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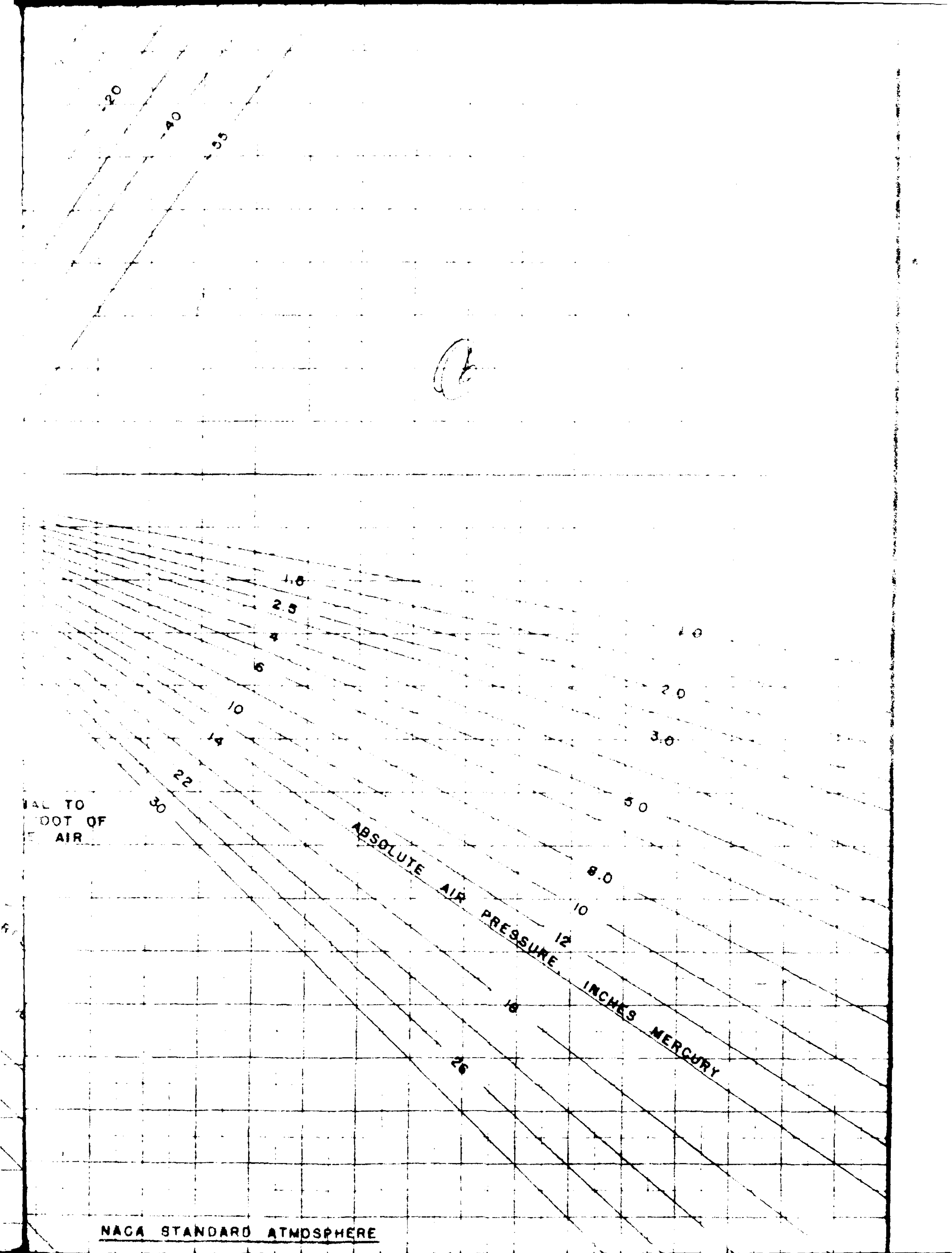
ABSOLUTE AI
Φ_3 IS PROPORTIONAL TO THE SQUARE ROOT OF THE ABSOLUTE AIR PRESSURE.
Φ_3
100
80
60
IDE 1000-FT
NACA STANDARD ATMOSPHERE



-20
-40
-55
1.6
2.5
4
6
10
14
22
30
1.0
2.0
3.0
5.0
8.0
10
12
18
26
ABSOLUTE AIR PRESSURE, INCHES MERCURY
TO
OF
AIR
NACA STANDARD ATMOSPHERE

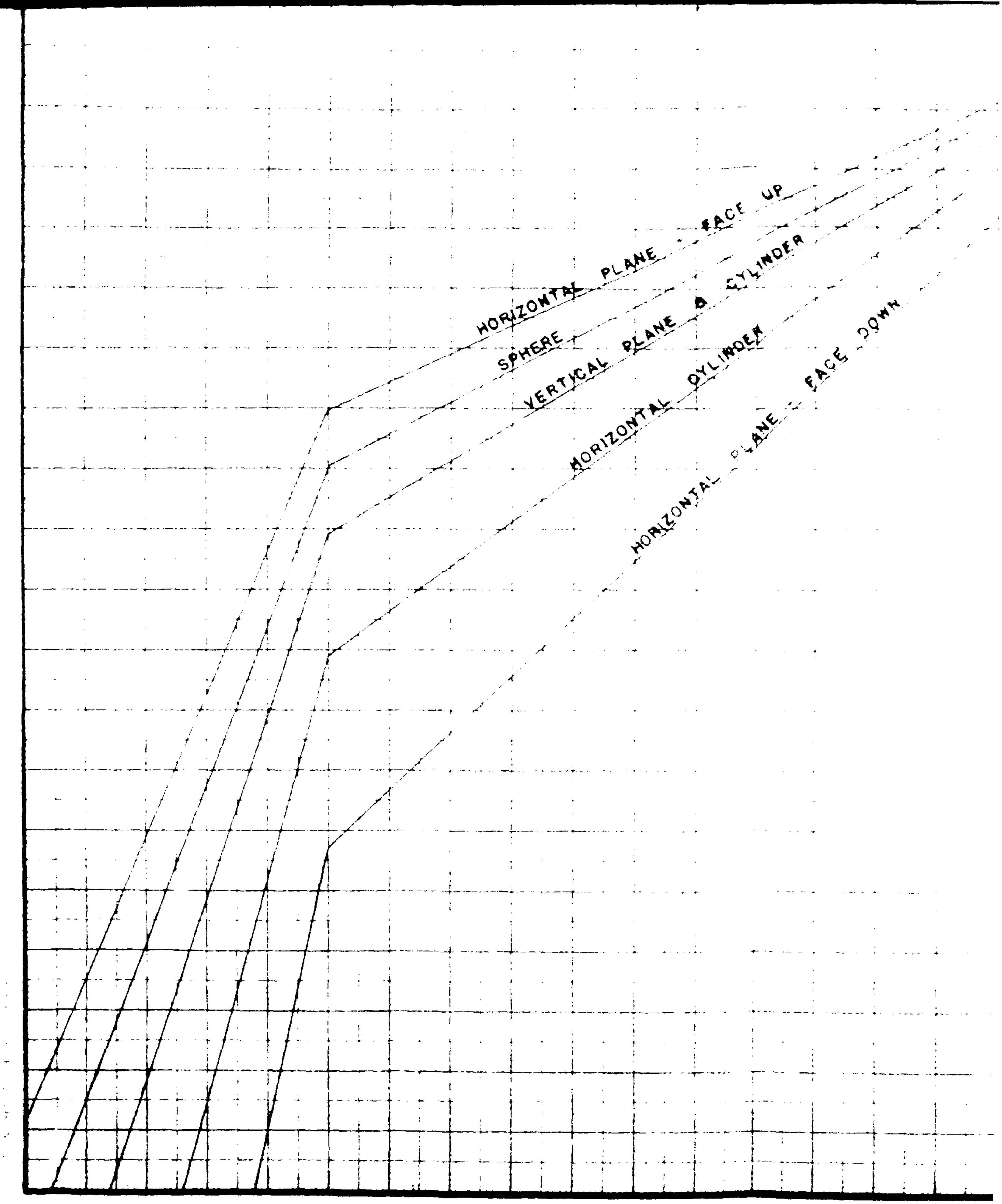


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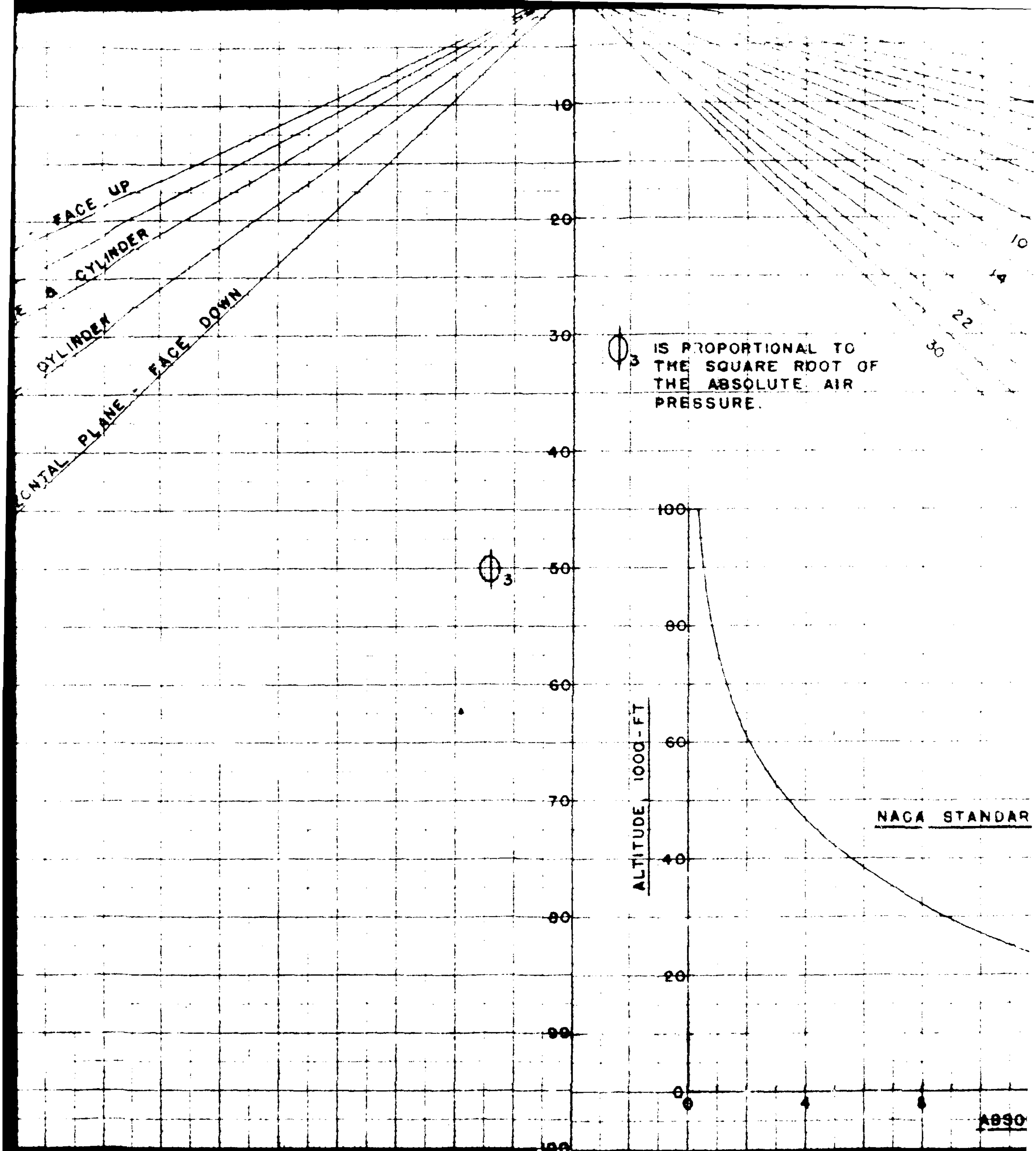
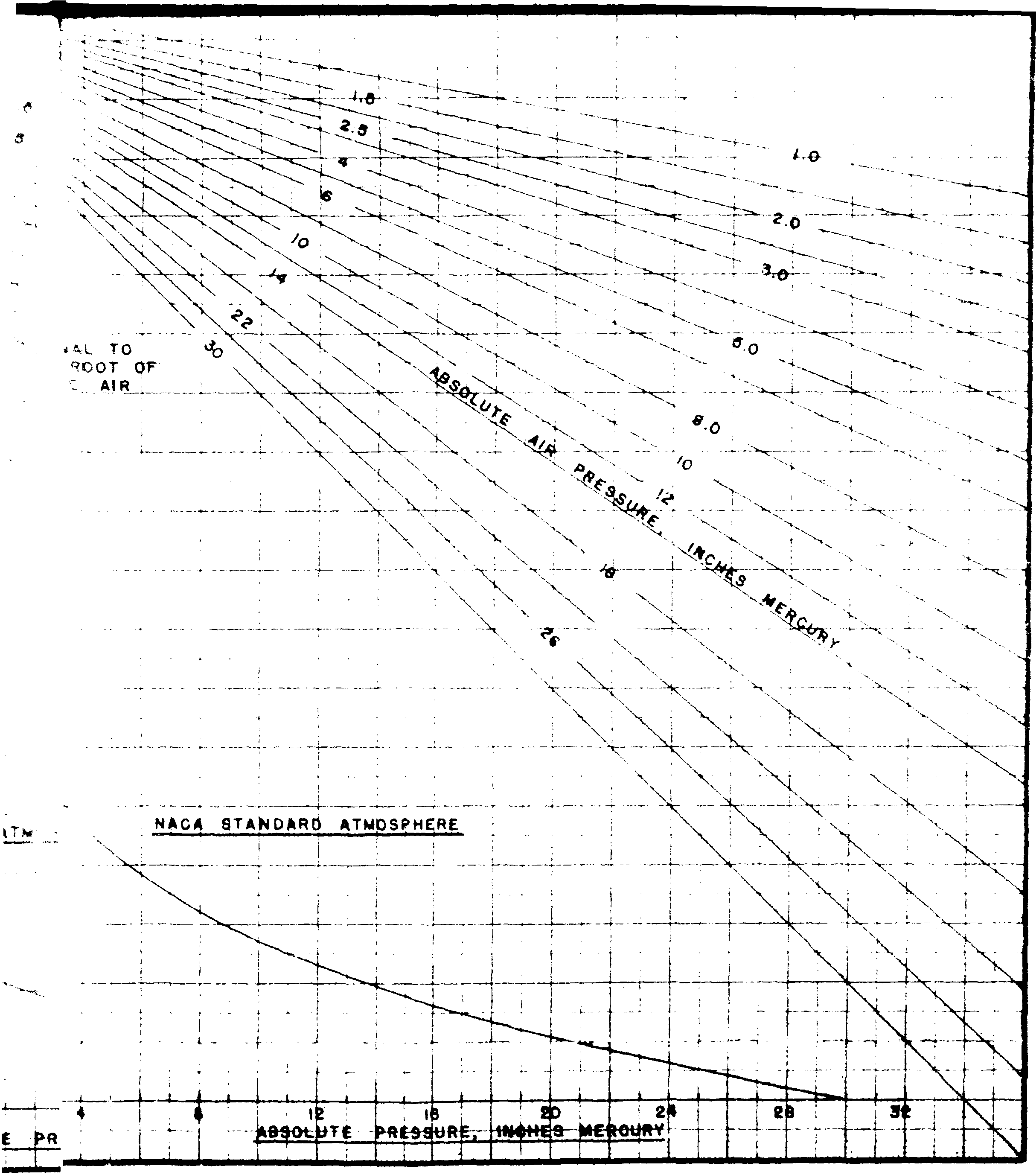
FIGURE II CHART FOR

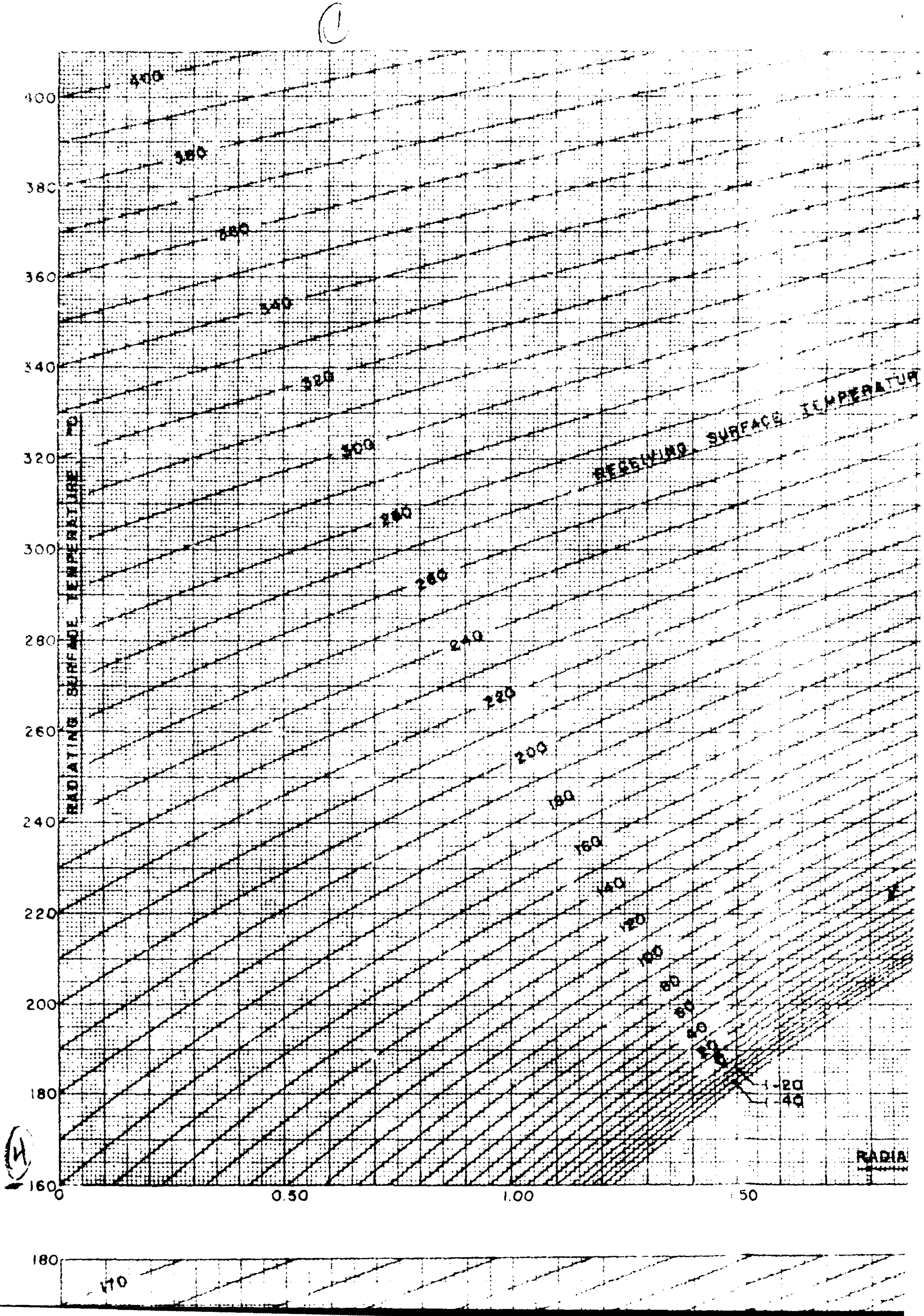


GURE 11 CHART FOR CALCULATION OF FREE CONVECTIVE HEAT DISSIPATION I

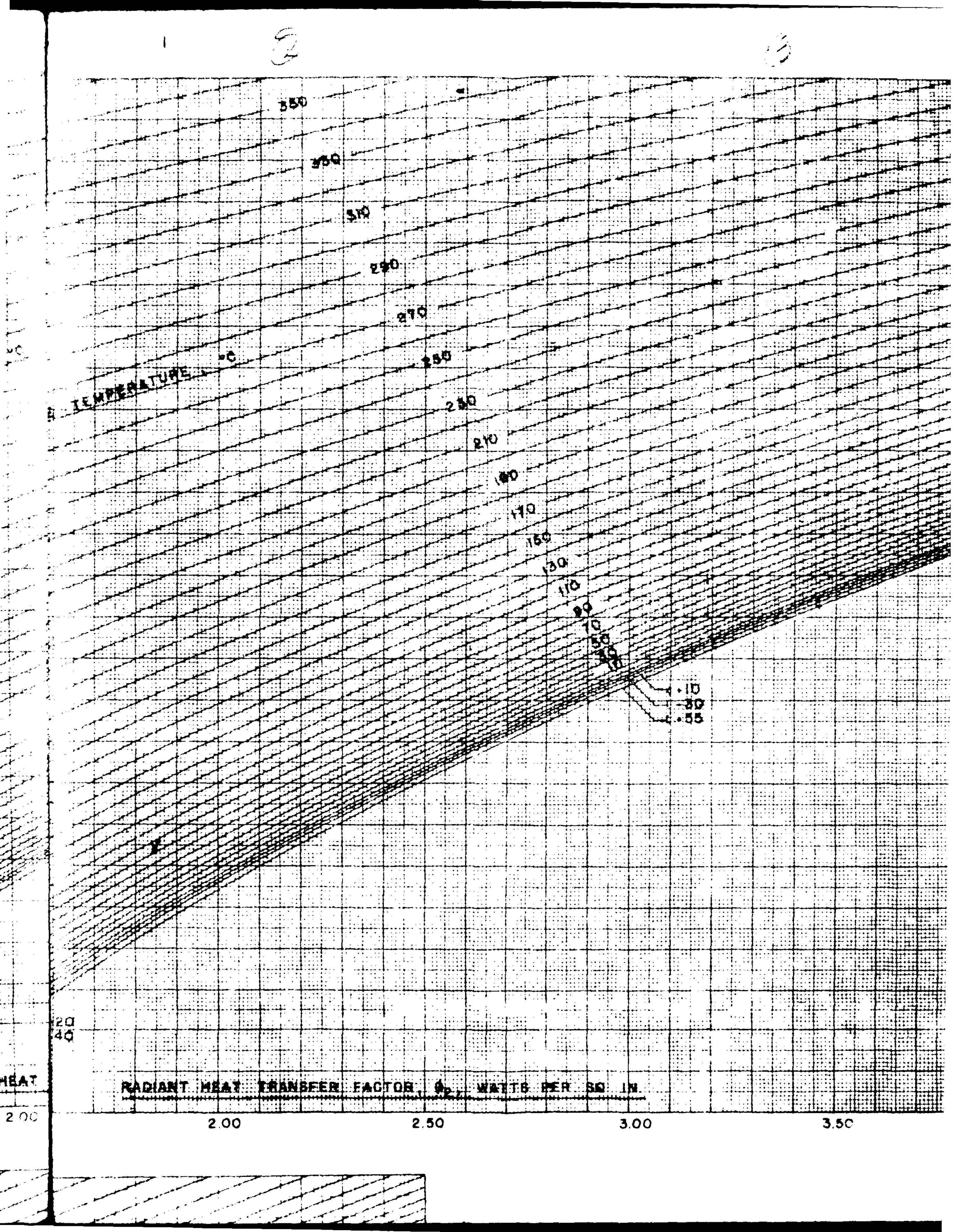
2



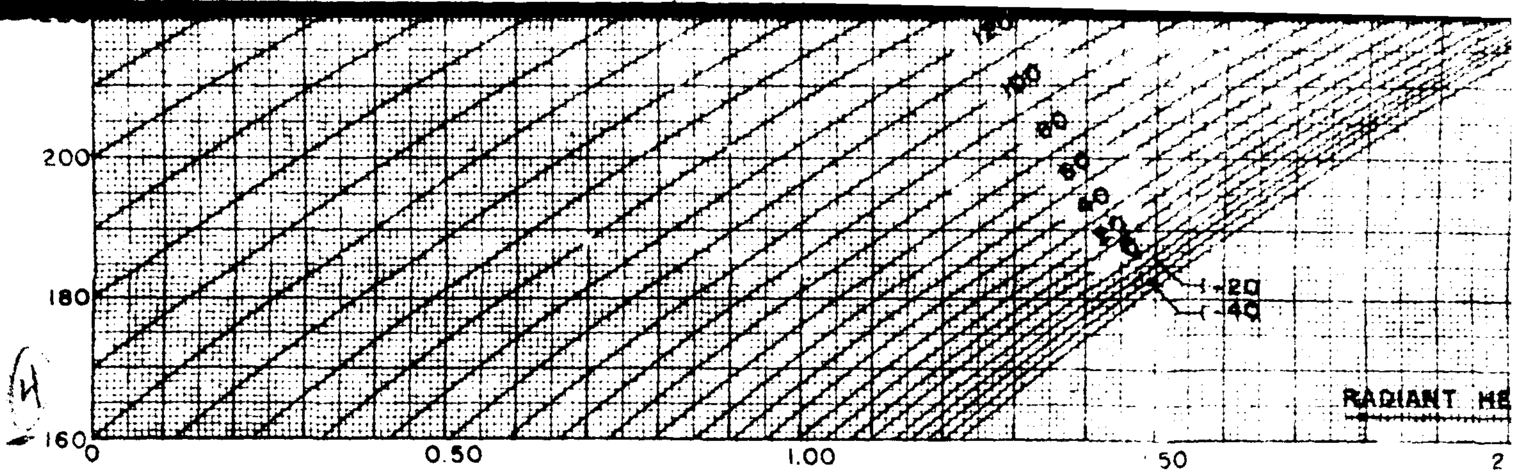
AIR HEAT DISSIPATION IN AIR

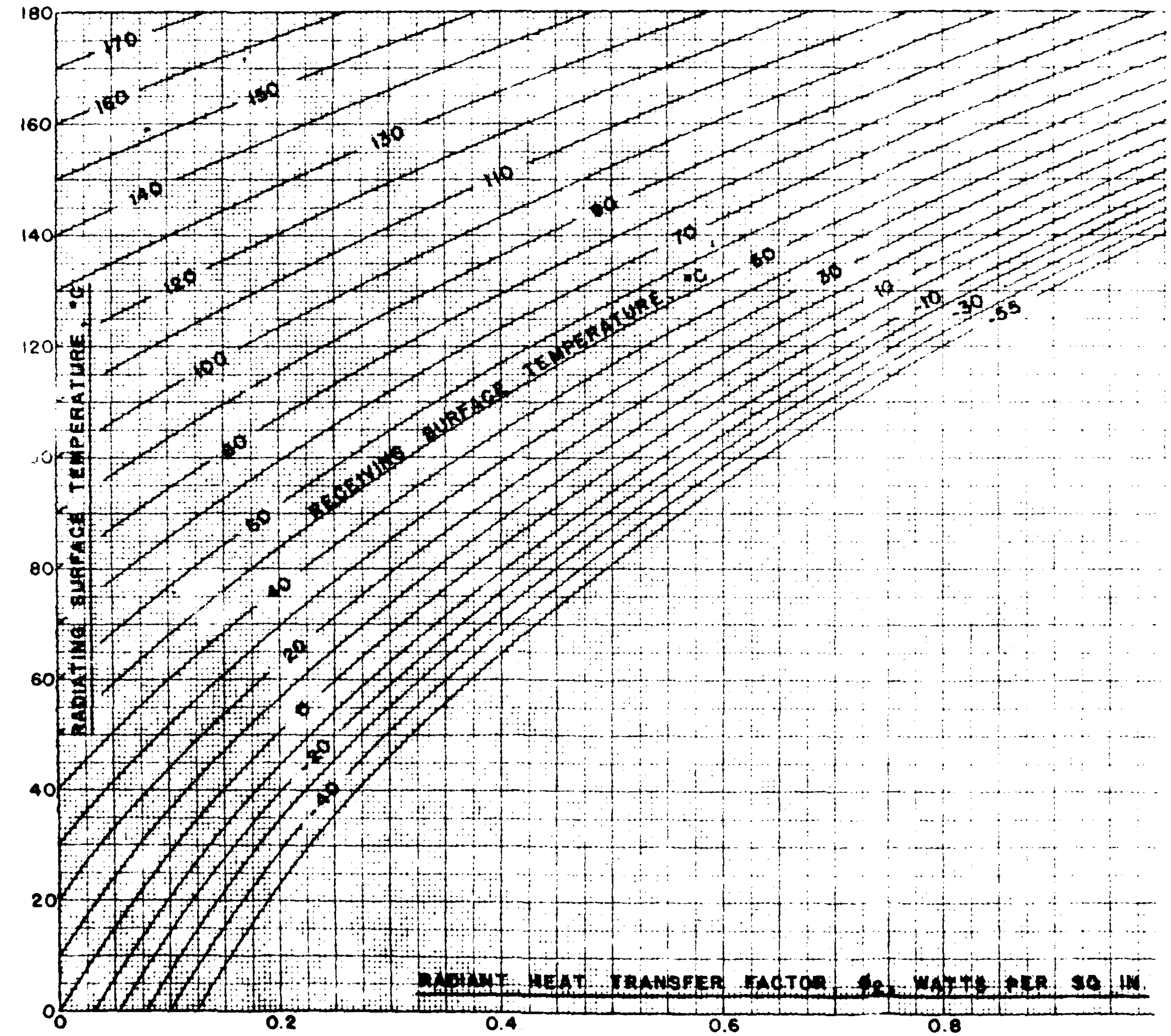


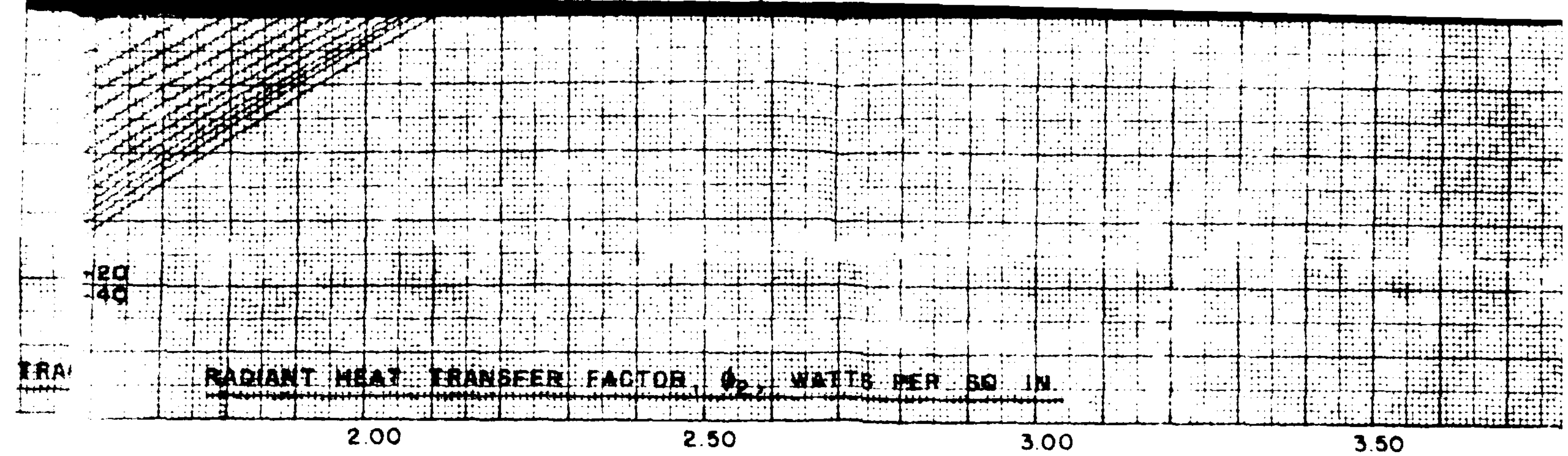
RADIATING SURFACE TEMPERATURE °F
RECEIVING SURFACE TEMPERATURE
400
380
360
340
320
300
280
260
240
220
200
180
160
0
0.50
1.00
1.50
RADIA
-20
-40
170



350
330
310
290
270
250
230
210
190
170
150
130
110
90
70
50
30
10
-10
-30
-55
TEMPERATURE, °C
120
140
RADIANT HEAT TRANSFER FACTOR, q_r, WATTS PER SQ IN
2.00
2.50
3.00
3.50
HEAT
2.00


RADIANT HE


RADIATING SURFACE TEMPERATURE °C
RECEIVING SURFACE TEMPERATURE °C
RADIANT HEAT TRANSFER FACTOR, WATTS PER SQ IN.



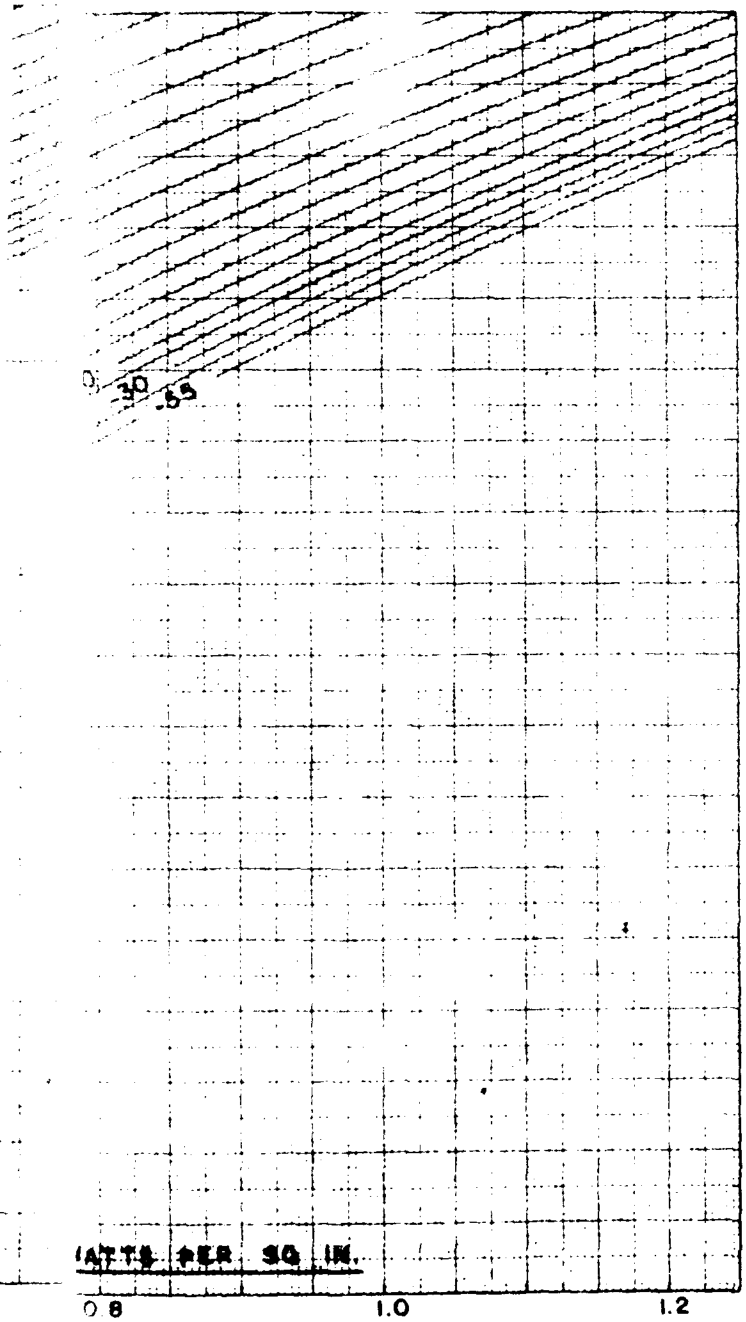


FIGURE 12 12 a (above)

12 b (left)

CHARTS FOR CALCULATION OF RADIANT HEAT DISSIPATION

BLACK BODY RADIATION

$\left.\begin{array}{l}\epsilon_1 = 1.00 \\ \epsilon_2 = 1.00\end{array}\right\}$ $F_e = 1$

CONFIGURATION $F_a = 1$

from AFTR-65

5

6